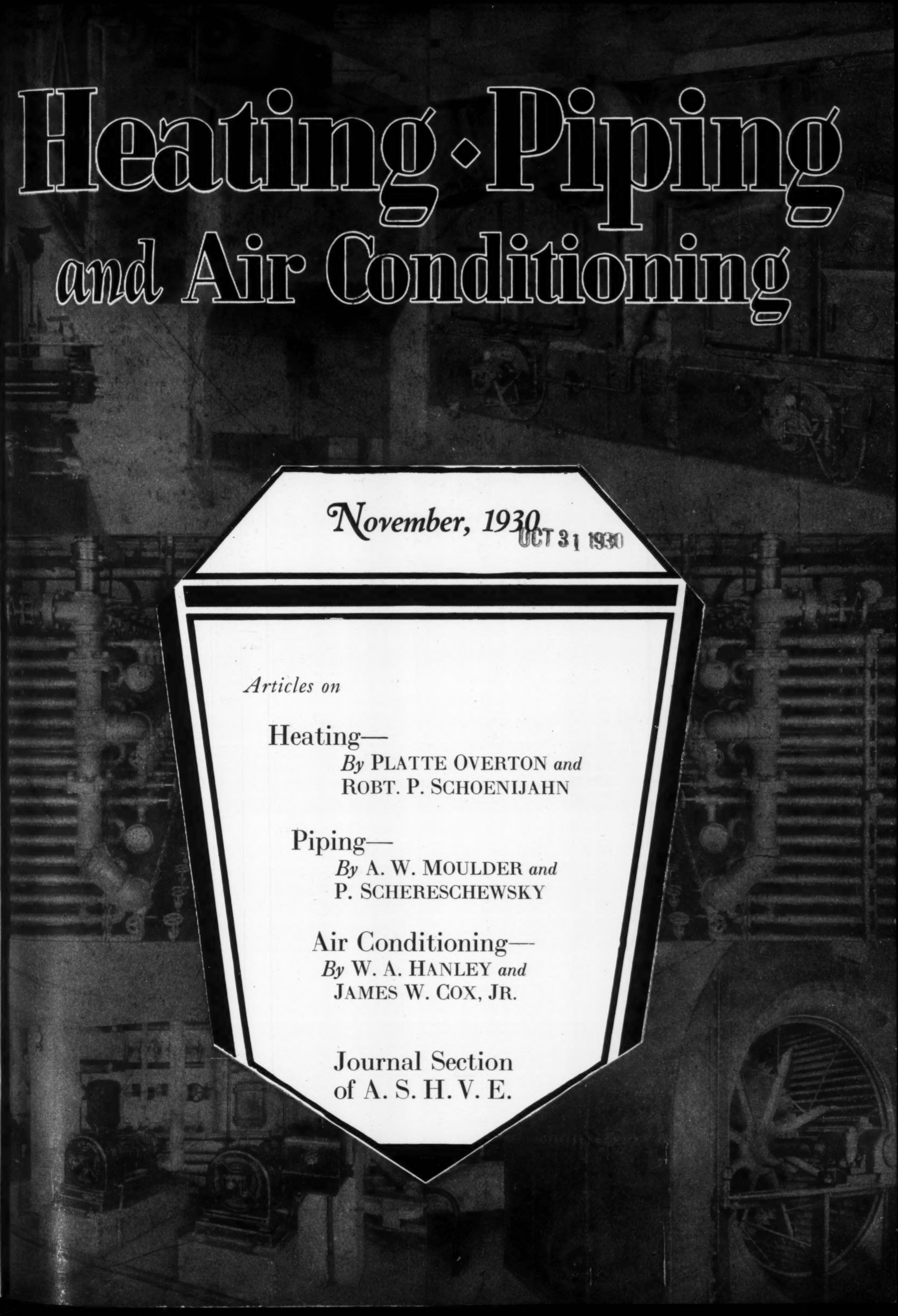

Heating · Piping and Air Conditioning

November, 1930

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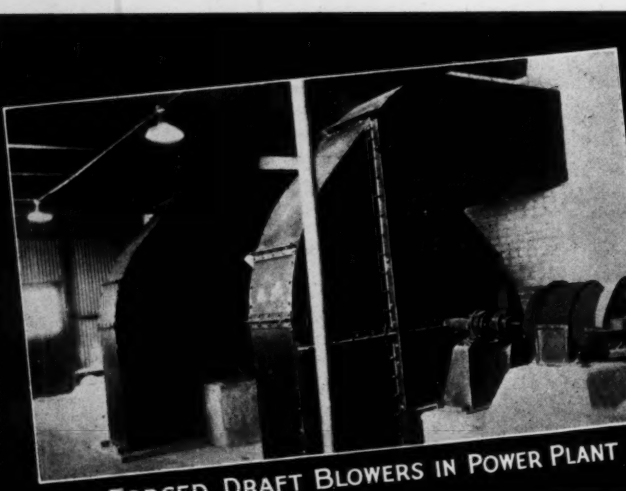

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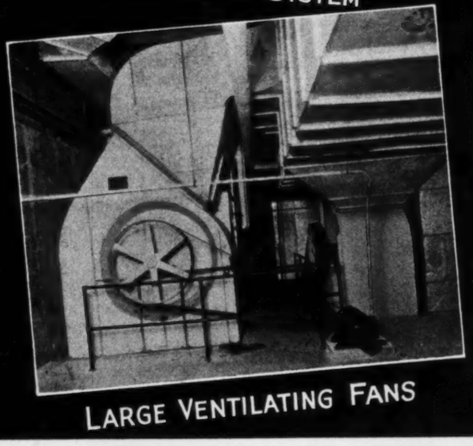

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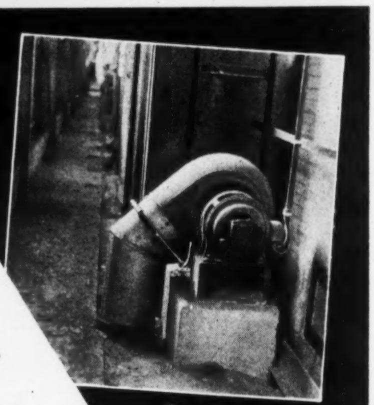

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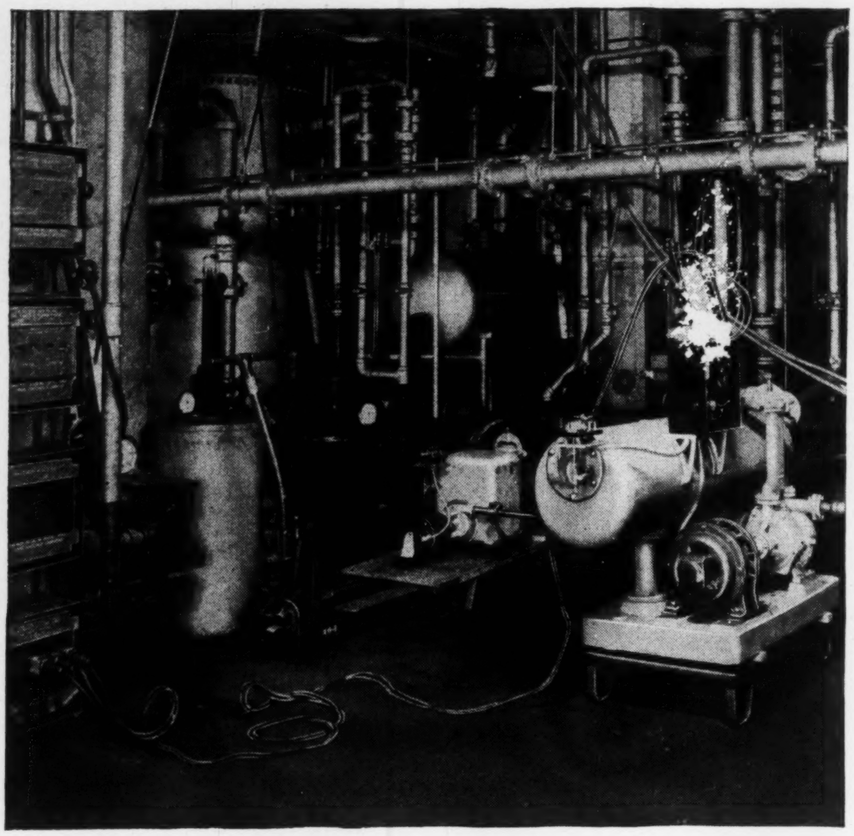

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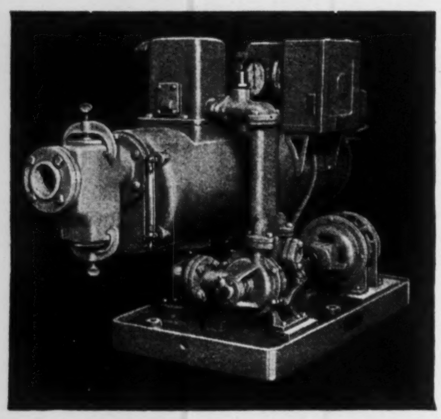

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Heating · Piping
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Vol. II November, 1930 No. 11

Mechanical Equipment of New Y. M. C. A. Building Typical of Best Practice

By Robert P. Schoenijahn*

TO MANY persons a Y. M. C. A. building may signify the local home of a national organization devoted to the physical, moral and spiritual upbuilding of American youth. Those entrusted with the administration of the modern Y. M. C. A. organization are charged with the operation of an institution incorporating the features of a hotel, a home, social center, athletic club, church and school. Provision for all of these activities become important factors in the planning of a new Y. M. C. A. building and the mechanical equipment of the institution necessarily presents many interesting and unique problems for the engineer engaged in the design of the heating, air conditioning, piping, plumbing and electrical work.

Basically, these problems require the same high grade of material and equipment that should be used in any structure when permanency is desired. But, at the same time, the element of cost must be most carefully considered, as the value of the various branches of mechanical, sanitary and electrical work have a well established ratio to the total building cost, as determined from hundreds of Association structures throughout the country. This information has been accumulated for a number of years by the architectural service bureau of the national Y. M. C. A., whose staff supervises many of the larger structures, and inspects the buildings after they have been in service, to record the relative value of materials and systems in use. These records serve as a valuable guide for other buildings.

FIG. 1—THE NEW Y. M. C. A. BUILDING AT WILMINGTON, DELAWARE. ITS MECHANICAL EQUIPMENT IS TYPICAL OF GOOD PRACTICE FOR THIS TYPE OF STRUCTURE

---

* Consulting Engineer, Wilmington, Delaware.

New Y. M. C. A. Building at Wilmington, Delaware

The community of Wilmington, Delaware, is quite proud of its new Y. M. C. A. building, completed several months ago, which was designed and supervised by Brown & Whiteside, Architects, Inc., of Wilmington, Delaware, co-operating with the architectural service bureau of the Y. M. C. A., retained by the local building committee to furnish technical data and details peculiar to this problem.

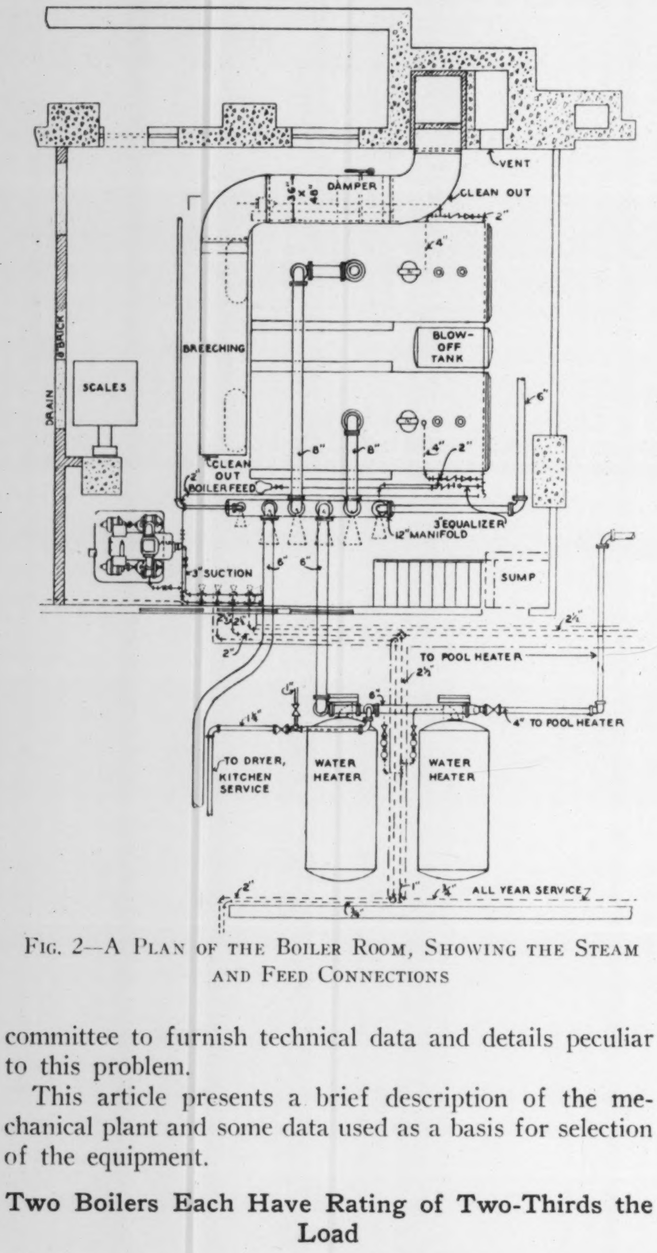


Fig. 2—A Plan of the Boiler Room, Showing the Steam and Feed Connections

This article presents a brief description of the mechanical plant and some data used as a basis for selection of the equipment.

Two Boilers Each Have Rating of Two-Thirds the Load

The boiler plant, illustrated in Fig. 2, has two steel boilers, each of 1,300 sq. ft. of heating surface, set extra high on brick foundations to properly accommodate motor-driven underfeed stokers, which are equipped with special electrically-controlled automatic pressure regulators, to operate the variable speed 220 volt, 3 phase, 60 cycle motors driving the fan and stoker. The regulation is sensitive in the regulation of steam pressure to within 1 lb. gage. The boilers operate without emitting smoke, on bituminous coal averaging 14,000 Btu, 20 per cent volatile and 6 to 8 per cent ash. Each stoker has approximately 36 sq. ft. gross area and is capable of operating continuously at 125 per cent of normal rating and at 150 per cent rating for a 3-hour period without cleaning fires.

The usual practice in Y. M. C. A. buildings is to provide a battery of three boilers, each having a capacity of one-third of the combined demand for house heating and for domestic hot water service. One boiler of one-third the total demand usually meets the summer requirements for hot water heating.

In this instance, space conditions limited the plant arrangement to two boilers, each having a normal rating of two-thirds of the combined total demand for heating and hot water purposes. One boiler with stoker equipment is normally used and has met the peak load conditions when operating at about 130 per cent normal rating. A single boiler has been operating successfully under the reduced summer demand with only one hot water generator in service. The best possible insulation was provided for the boilers, breeching, equipment, tanks, piping, etc., in order to reduce the losses from radiation and to provide a comfortable boiler room during warm weather.

Each boiler is completely fitted with the usual accessories, including an illuminated water column, and an automatic water feeder with low water electric cut-out connected to stoker control. Each blow-off is provided with a blow-off valve connected to a header discharging into the blow-off tank, which is drained to a sump pit.

The Boiler Room Piping

The pipe work in the boiler room was carefully detailed; each boiler is connected with an expansion bend to a 12-in. welded steel manifold having flanged nozzles for each of the principal branch mains, which are separately valved. One branch main for "all year" service supplies steam to hot water generators, pool heater, dry room heaters and to the dishwashers in the kitchen. The branch mains of the heating systems are circuited and valved to permit control of different sections of the building, as it is often desirable to provide heat in mild weather in physical department or social rooms, when no heating is required in other sections of the building.

All the return mains for each of the corresponding steam lines are connected into a header, so valved that each branch can be cut-off and be separately drained to the sewer; also a parallel header is provided with cross-connections valved to permit condensate to be returned to the boiler through vacuum pumps or by gravity, or to be drained to the sewer, as may be required, in an emergency. These features were particularly valuable during the months of temporary heating and when the system was being cleaned. Provision is made by special connection for the measuring of condensation, in case of conducting an evaporation test.

Vacuum System Used

As may be inferred from this description, a vacuum system of heating was installed, with the view of securing rapid circulation to the most remote radiators and particularly to eliminate noise in dormitory sections.

The system includes a self-contained duplex pumping unit, each pump having a rated capacity of 26,000 sq. ft. E. D. R. The pump is connected with the heating return header and arranged, normally, to return condensate to the boiler, or it may pump direct to the sewer in emergency. The piping system, a year after its installation, still maintains a vacuum from 7 in. to zero pounds for periods of 15 minutes during light load conditions.

Estimating Boiler Demand

In arriving at the total boiler demand, the requirements for house heating, hot water service and heating of the pool water were carefully determined. The house heating demand included the direct radiation, to which was added the equivalent direct radiation of the various ventilating heater units delivering out-door air into the building. This total radiation demand is approximately 15,000 sq. ft. E. D. R.

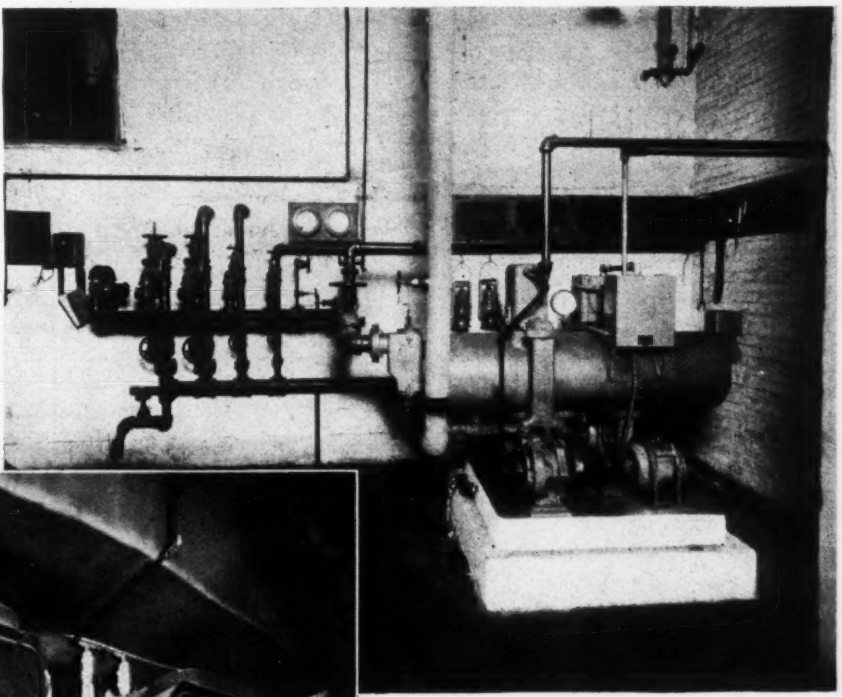


FIG. 4—THE VACUUM PUMP FOR THE HEATING SYSTEM, SHOWING MANIFOLD SUCTION AND DISCHARGE HEADERS WITH SEPARATE CONTROL VALVES ON EACH BRANCH RETURN

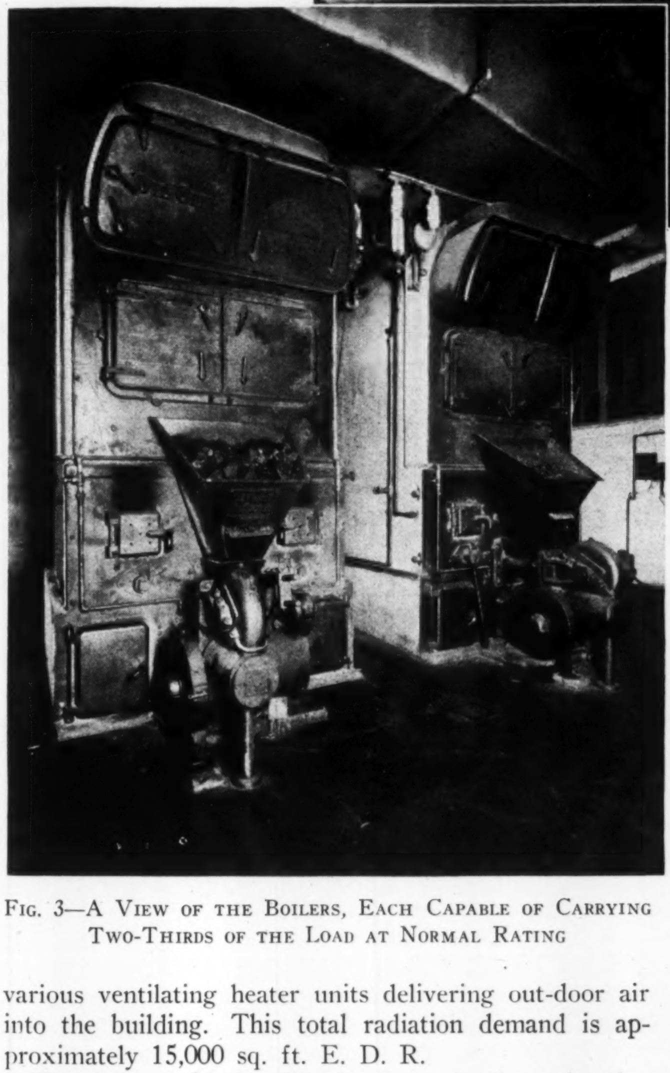


FIG. 3—A VIEW OF THE BOILERS, EACH CAPABLE OF CARRYING TWO-THIRDS OF THE LOAD AT NORMAL RATING

The demand for heating the water in the swimming pool, when water is being re-circulated, is comparatively light, and the heating of raw water when refilling the pool can and is usually done during the night and on Sundays, when the demand for domestic hot water is at a minimum; hence, the steam requirement for heating the pool water was not considered in determining the size of boilers.

Experience and data gathered from many Y. M. C. A. buildings has developed a practice of determining domestic hot water requirements on the basis of a peak demand for heating 100 gallons of water from 50 to 180 F per hour, for each shower in the physical department. This value has proven satisfactory and not excessive. As the dormitory showers and other demands for domestic hot water do not coïncide with the peak demand of the physical department, these requirements are ignored.

Two hot water generators are provided, each having a steam coil sufficient for the peak demand noted above, and each tank having a storage capacity of 50 per cent of the peak demand. Automatic water temperature control valves are essential for good service. This installation included two heaters, 54 in. × 120 in. long, having a storage capacity of 1,190 gallons, with return bend copper heating surface sufficient to raise 2,000 gallons of water from 40 to 180 degrees with steam at 1 pound gage.

Heater for Swimming Pool

The heater for heating the swimming pool water is of the instantaneous two-pass type, having a capacity to heat 5,000 gal. of water per hour from 40 to 80 F. This is normally used to maintain the pool at 72 F by heating the recirculated, filtered water. When filling the pool with fresh city water, the supply is taken from the house heaters and then through the pool heater in order to accelerate the time required for filling and heating.

When the plant was first put in operation both tank heaters were inadvertently placed in service. Each tank is equipped with an 8-in. dial, 8-day recording thermometer. To effect economy in fuel, one of these tanks was cut out of service and the recording chart produced an interesting graph illustrating the value of good insulation of the hot water tank. This is reproduced in Fig. 5 and shows that the temperature of the water in the tank dropped from 145 F to 107 F with fairly uniform increments in seven consecutive days.

Installation of Steam Piping

All steam piping was installed with standard weight steel pipe, with standard weight cast iron fittings and flanges, flanged fittings being used at connections to the header and at certain points in the mains to facilitate installation; otherwise screwed fittings were used.

Certain of the return mains were, of necessity, run below the basement floors, in which case extra heavy wrought iron pipe was used with wrought line sockets and fittings, the connections being welded at all joints over exposed thread and fitting. These lines were installed in split terra cotta tile with roller supports. The runouts to risers were built into concrete trenches and arranged with swing joints to compensate for expansion.

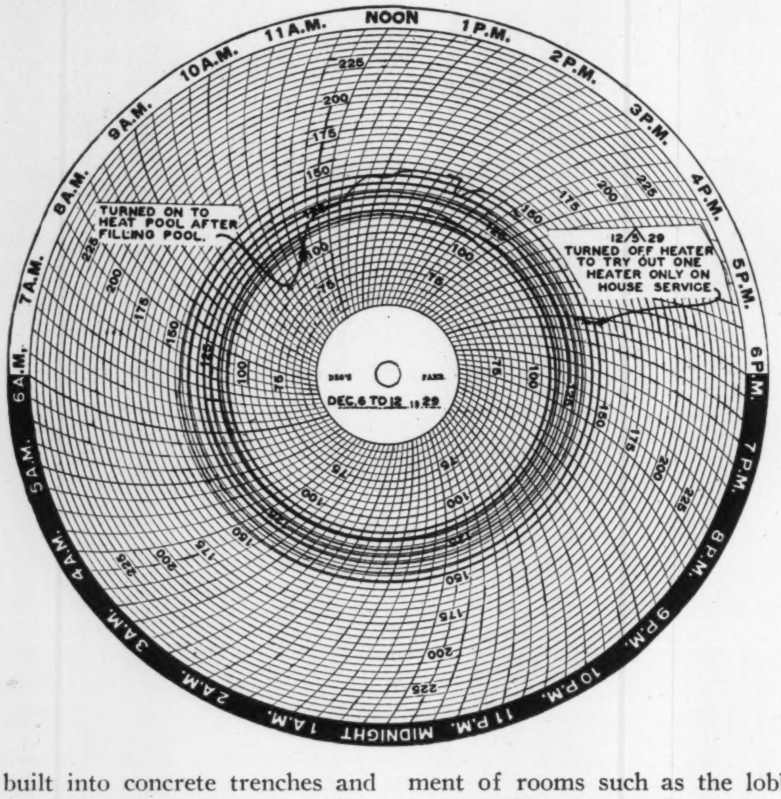


Fig. 5 — Results of a Test on the Heaters for the Pool Water

All pipe work was tested with hydraulic pressure to 100 lb. gage to detect any poor fittings or imperfect threads. This work was carefully inspected and for the size of the operation and conditions encountered, very few defects developed, though several bad fittings and a few split pipes were found and replaced. This testing and supervision is all the more important where many of the mains and all the risers are concealed within building construction, as in this case.

Where practical, turns and offsets were made with pipe bends to eliminate unnecessary fittings. Branches and runouts from horizontal mains to risers are assembled with three-fitting swing joints and all risers passing through floor construction have pipe sleeves of ample diameter to receive the insulation, that it may be continuous. As the building is but six stories in height, the risers were anchored at the third floor level.

Runouts from risers are installed in the furred space over the window heads and in each case serve the radiator on the floor above.

Dripping the Mains

Owing to structural difficulties, an unusual number of low points developed in the supply mains; this required special study to properly drip mains and still have access to the valves, strainer and trap combination, provided at each such drip point. In a few instances, short branch mains were dripped by double service valves through a radiator at the end of the run.

Radiation

Throughout the dormitory section and work spaces, cast iron radiation with extra high legs is installed exposed under windows, each equipped with a hand control valve. In the sections of the building with special treatment of rooms such as the lobby, social halls, library, lounge, billiard and game rooms, the radiation was concealed in cabinet work, using cast iron radiation. Where space conditions were limited, extended surface copper radiation was used. All of these public spaces are provided with a dual system of automatic temperature control.

The Ventilating Units

The auditorium was equipped with special ventilating heater units, arranged to provide outside air in conjunction with automatic temperature control and pneumatically operated dampers to admit outside air, or to permit re-circulation of room air for quick heating-up, as this space is for the greater part of each day set to maintain a maximum of 50 F using the units on gravity. On forced recirculation, the units are capable of warming the room to 68 degrees on short notice.

A separate system of mechanical exhaust ventilation is used in connection with the heater units, arranged with exhaust ducts and pneumatically operated dampers to permit air to be removed either through grilles near the floor, or from near the ceiling during smokers or dinners.

Heating the Swimming Pool Room

The heating of the swimming pool space is always a problem, as the bathers demand a temperature close to 80 F and spectators naturally complain, as they are

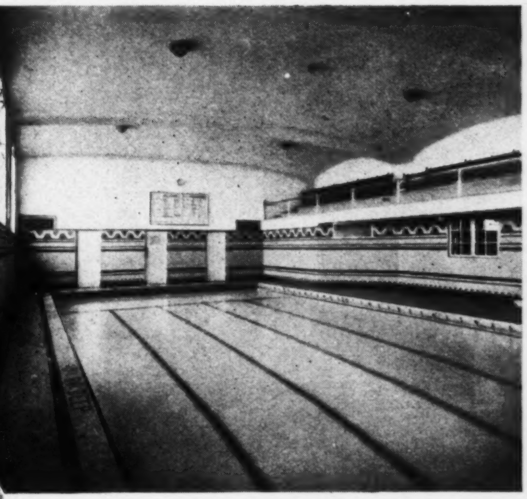


FIG. 6—THE POOL ROOM IS HEATED BY VENTILATING HEATER UNITS DELIVERING OUTDOOR TEMPERED AIR THROUGH GRILLES AT THE END AND EXHAUSTED THROUGH VENT GRILLES AT THE SIDE AND IN THE DOORS.

FIG. 7 — THE SWIMMING POOL FILTER AND PUMP ROOM

clothed and are usually in a gallery. Excessive humidity and "pool" odors also add to the discomfort.

In this instance, an attempt to better these conditions resulted in the installation of ventilating heater units delivering tempered outdoor air through grilles in the end walls, set about 8 ft. 6 in. above the floor to avoid draft on the bathers. A small amount of extended surface radiation in copper casings, concealed in tiled walls, is placed under each of the large windows, sufficient to offset the glass losses and prevent downdraft.

The air introduced into the room represents about a 10-minute air change, all of which is exhausted through the grilles by the mechanical exhaust system, through the locker rooms, shower and dry rooms where it is desirable to maintain the same temperature as in the pool. This arrangement materially reduces the excessive humidity usually found in these departments.

Temperature Control

The pool radiation is connected with a dual system of thermostatic temperature control so arranged that the tempered air is delivered at about 79 F; a separate thermostat controls the gravity heater units so these will come into service if the room temperature falls to 78 F and shut-off at 79 F. Pneumatically operated dampers are placed in the outdoor air intake ducts, which are normally closed and are opened through an electrical switch when the heater fan motor is started. All the units are controlled from a central point and are in charge of the pool attendant who has the control of the exhaust fan system serving the locker, shower and dry-rooms in this department.

The construction and space conditions in the building made it very difficult to install supply and return risers for direct radiation in the main and auxiliary gymnasiums and the window spacing and height of the sills precluded the proper placing of such radiation without unsightly arrangement and poor distribution. Hence the ventilating heater units solved a difficult problem. Four units of 1,500 c.f.m. were installed to deliver outdoor air, and three heaters of 1,200 c.f.m. were used as recirculating units to aid in quick "heating up." All of these units are connected with the dual system of thermostatic temperature control similar to the swimming pool.

A recirculating type heater unit also solved the problem of heating for the cafeteria, which is of Spanish design with high arched windows and casement doors leading out to a terrace, leaving no space for radiation, nor could gravity indirects be installed, as this room had no cellar or excavation below. Exposed radiation would not be in keeping with the Spanish design, so a recirculating heater unit placed in a tunnel on the side wall provides for heating and creates air circulation.

A separate system of mechanical exhaust is provided for the kitchen and at the service counter of the cafeteria. Low ceilings in these two spaces make it difficult to ventilate even with a 1½ minute air change.

An electric power plant in the Southwest recently expanded its facilities, necessitating replacement of considerable piping with lines of larger diameter. Alteration of the low pressure exhaust main was by far the most intricate part of this job. The piping contractor had to tear out the old 14-in. diameter line and install a 24-in. exhaust main from the electric generating units, in an extremely cramped space. The old pipe was removed with oxy-acetylene cutting blowpipes.

It was necessary to provide two 14-in. outlets on the 24-in. main, one at the top and one at the side connecting with a relief valve. Provision for the outlet for the latter connection was complicated by the fact that the relief valve was connected with a 14-in. line which had to pass over the 24-in. main and at the same time clear a concrete beam projecting from above. The result was that the 14-in. outlet in the side of the 24-in. main had to be offset below the center line.

Openings of the proper size and shape were cut according to templet in the 24-in. main, using the cutting blowpipe. Two short lengths of pipe, each having a flange on one end, were then prepared by cutting the unflanged end according to a templet so that it would fit the opening in the main. After beveling the edges of the openings and of the nozzles, the two nozzles were welded in turn to the 24-in. main.

Economical Steam Pipe Sizes *for* Heating, Process and Power Steam

Article No. 4

By **A. W. Moulder**

This is the fourth installment of the series on steam pipe sizes; starting in the August HEATING, PIPING AND AIR CONDITIONING, and published in consecutive months, this series forms a valuable reference for any engineer who wishes to obtain the balance between the theoretical and practical in piping design, both high and low pressure.

IN THE preceding three examples the attempt has been made to show how this method of pipe size calculations may be used in a practical manner, yet with results approaching theoretical perfection when applied to most types of low pressure heating work. This next step will be to show how the same tables and the same general methods may be applied to any ordinary steam pressure and temperature. Typical calculations are shown in detail.

By referring back to the formula from which the tables are prepared (see the August HEATING, PIPING AND AIR CONDITIONING) it will be noted that the only "variable" in the formula is the density of steam corresponding to the pressure and temperature under which the piping is to operate. Therefore, it follows that the weight of steam carried in a pipe of given size when operating at any pressure other than that upon which the tables are based (½ lb.) will vary directly as the difference in density affects the formula. In other words, for any given steam pressure and temperature a factor can readily be developed which will express the change in density in the terms of effect upon the pounds weight of steam carried in a pipe of given size and with a given loss in pressure.

Conversion Factor Tables

The tables of conversion factors have been developed for this purpose. With these two tables the pounds weight of steam as given in the pipe size tables may be corrected readily for any steam pressure and temperature which may be encountered. At increased pressures and temperatures these factors are used by multiplying the pounds weight of steam carried as given in the steam pipe size tables by the factor under the pressure and temperature under consideration. The following simple example will serve to illustrate not only the solution of the specific problem presented but also the proper use of the conversion factors.

FACTORS FOR CONVERSION * OF LOW PRESSURE STEAM PIPE SIZE TABLES TO MAKE THEM APPLICABLE TO HIGHER PRESSURES AND TEMPERATURES

PRES. LB. GAGE	SATURATED STEAM	SUPERHEAT—F.								
		20	50	100	150	200	250	300	350	400
½	1.00	.99	.96	.93	.90	.87	.85	.83	.81	.79
5	1.15	1.13	1.10	1.07	1.03	1.00	.97	.95	.93	.90
10	1.27	1.25	1.23	1.19	1.15	1.12	1.09	1.06	1.03	1.01
15	1.39	1.37	1.34	1.29	1.25	1.22	1.18	1.15	1.13	1.10
20	1.49	1.47	1.44	1.39	1.35	1.31	1.27	1.24	1.21	1.18
25	1.59	1.57	1.53	1.48	1.43	1.39	1.36	1.31	1.28	1.25
30	1.69	1.66	1.62	1.57	1.52	1.48	1.44	1.40	1.37	1.34
35	1.77	1.74	1.71	1.65	1.60	1.55	1.51	1.47	1.44	1.40
40	1.85	1.82	1.79	1.72	1.67	1.62	1.58	1.54	1.50	1.47
45	1.93	1.90	1.86	1.80	1.74	1.69	1.65	1.61	1.57	1.53
50	2.01	1.98	1.93	1.87	1.81	1.76	1.71	1.67	1.63	1.59
60	2.14	2.11	2.06	1.99	1.92	1.88	1.83	1.78	1.74	1.70
70	2.28	2.24	2.19	2.12	2.05	1.99	1.94	1.89	1.85	1.81
80	2.40	2.36	2.31	2.23	2.16	2.10	2.05	1.99	1.95	1.90
90	2.51	2.47	2.42	2.34	2.27	2.20	2.14	2.09	2.05	2.00
100	2.63	2.58	2.53	2.44	2.37	2.30	2.24	2.18	2.13	2.08

*At increased pressures and temperatures steam pipe will carry load as indicated in low pressure table times (x) conversion factor as above with drop per lineal ft. as indicated in low pressure table.

To simplify the use of above tables it is suggested that the steam loads on the job in question be corrected by dividing same by proper factor above. Then apply corrected load directly to low pressure table.

FACTORS FOR CONVERSION * OF LOW PRESSURE STEAM PIPE SIZE TABLES TO MAKE THEM APPLICABLE TO HIGHER PRESSURES AND TEMPERATURES

PRES. LB. GAGE	SATURATED STEAM	SUPERHEAT—F.								
		20	50	100	150	200	250	300	350	400
125	2.88	2.84	2.77	2.68	2.60	2.52	2.46	2.40	2.34	2.29
150	3.12	3.07	3.00	2.90	2.80	2.72	2.65	2.59	2.53	
175	3.33	3.28	3.20	3.09	3.00	2.91	2.84	2.77	2.71	
200	3.54	3.48	3.40	3.28	3.18	3.09	3.01	2.94	2.87	
225	3.72	3.67	3.58	3.45	3.34	3.25	3.17	3.09	3.03	
250	3.88	3.85	3.75	3.62	3.51	3.41	3.32	3.25		
275	4.08	4.02	3.92	3.78	3.67	3.56	3.47	3.38		
300	4.26	4.19	4.08	3.93	3.81	3.70	3.61	3.51		
350	4.58	4.50	4.38	4.22	4.09	3.97	3.87	3.77		
400	4.88	4.79	4.67	4.49	4.35	4.22	4.11	4.02		
500	5.43	5.32	5.18	4.99	4.82	4.68	4.56			
600	5.95	5.82	5.66	5.45	5.26	5.10	4.97			
900	7.29	7.12	6.91	6.60	6.37					
1200	8.54	8.29	8.00	7.60	7.30					
1350	9.15	8.84	8.50	8.06						

*At increased pressures and temperatures steam pipe will carry load as indicated in low pressure table times (x) conversion factor as above with drop per lineal ft. as indicated in low pressure table.

To simplify the use of above tables it is suggested that the steam loads on the job in question be corrected by dividing same by proper factor above. Then apply corrected load directly to low pressure table.

EXAMPLE NO. 4— COMBINATION PROCESS PIPING AND HEATING SYSTEM

Referring to Example 1 (in the September issue) it will be assumed that steam is to be supplied to that same low pressure heating system through a reducing pressure valve from a boiler plant some distance away operating at 100 lb. saturated steam pressure. It will also be assumed that a certain amount of this high pressure steam is going to be used in the factory building (for which the heating system was designed) for drying or other process work. Fig. 6 illustrates the problem.

The problem, of course, is to calculate the economical yet practical pipe sizes for these steam lines. In the diagram:

- *W* represents the line supplying steam from tee *Y* to the pressure reducing valve, through which low pressure steam is admitted to the heating system.
- *X* represents the line supplying steam from tee *Y* to the dryer.
- *Z* represents the steam line from the boiler header (where steam is maintained at 100 lb. pressure) to the "common" tee *Y*.

Effect of Insulation

Process steam must be supplied to the dryers all year round, therefore it will be advisable to thoroughly insulate all of this high pressure piping. This is a logical conclusion because the surface of this piping has no value as a heating medium during the summer months and, furthermore, the heat given off during that period would be objectionable. With proper insulation, therefore, it may be assumed that the extra loss in pressure due to heat given off by the line by radiation may be neglected as a part of this problem.

First step: As in previous examples the first step will be to tabulate the pieces in the main piping which constitute the longest run and then record opposite these tabulations the calculations in the order as they are made. (The tables referred to in this and following computations are reproduced herewith in part if they have appeared in full in previous issues. Otherwise, they appear in full in this issue.)

Piece	Load	*Corrected	Length		Equiv.
Mark	Lbs.	Load, Lbs.	Ft.	Ells	Lgth.
Z	2345	977	260	7	309
X	700	292	70	6	100
		Totals	330	13	

*See the second paragraph following.

Determination of Maximum Load

To calculate the *minimum* sizes of pipe permissible the *maximum* loads which will be carried must be determined. This is especially necessary in meeting the minimum pressure requirements for the process steam at all times. The load for the dryer is constant after it is once in operation, therefore the maximum condensing load under normal operating conditions will be correct for all calculations. In this case the maximum dryer load has been placed at 700 lb. weight of steam per hour. With the heating system load, however, allowance must be made for the fact that the temperature of the building may be allowed to drop over night or week-ends to a point close to freezing—say 40 degrees. This (for a period of time) increases the steam demand for heating above the normal load estimated and allowance must be made for this so that the pressure to the dryers will not drop below that required. With a unit heater system this starting load is readily obtainable by reference to capacity tables of unit heaters and in this case the increase has been placed at 40 per cent above normal. If this were a radiator or coil system the increase should be greater (probably double normal) due to the greater weight of materials to be heated up during this "starting load" period.

The "Corrected Load"*

This "corrected load" represents a change to an artificial load obtained by use of the conversion factors to compensate for the change in steam pressure from that upon which the tables are based (½ lb.). In this example it is not advisable to calculate for as great a drop in pressure between the boiler header and the reducing pressure valve as might be used if steam were supplied to the heating system only. This is for the reason that steam is also supplied to the dryers which must have a maximum pressure at their inlet of from 60 to 70 lb. The formula for pressure drop is based upon the mean density of steam between the starting point and the finish, therefore it will be assumed that instead of 100 lb. steam pressure an average pressure in the line of approximately 80 lb. will be more nearly correct and the "conversion factor" for 80 lb. saturated steam will be used. By reference to the conversion factor table this is found to be 2.40.

The "corrected load" is consequently found by divid-

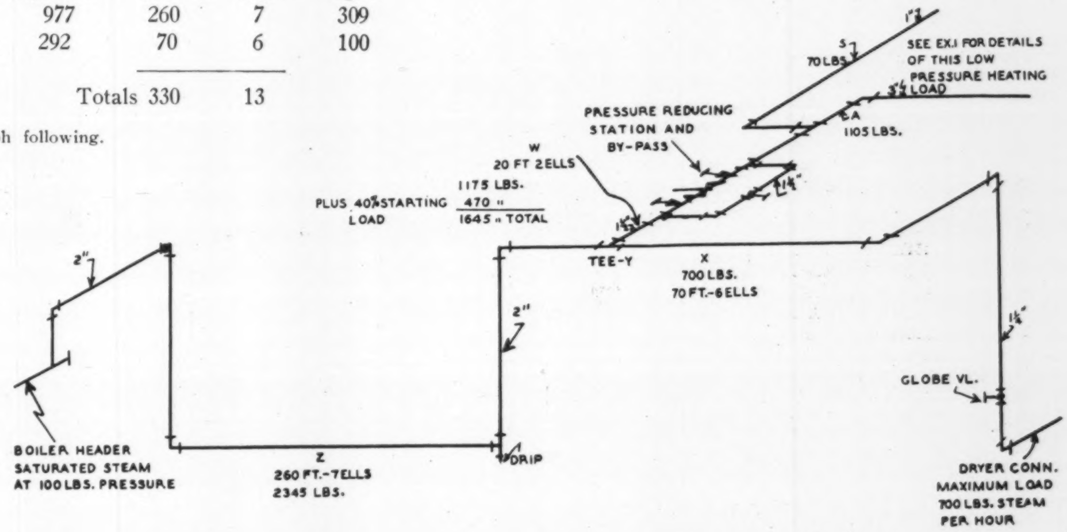

1"
S
70 LBS
SEE EX.1 FOR DETAILS OF THIS LOW PRESSURE HEATING LOAD
3½"
A
1105 LBS.
PRESSURE REDUCING STATION AND BY-PASS
W
20 FT 2 ELLS
1175 LBS.
PLUS 40% STARTING LOAD
470 "
1645 " TOTAL
1½"
1¼"
TEE-Y
X
700 LBS.
70 FT.-6 ELLS
2"
2"
1¼"
GLOBE VL.
BOILER HEADER SATURATED STEAM AT 100 LBS. PRESSURE
Z
260 FT.-7 ELLS
2345 LBS.
DRIP
DRYER CONN. MAXIMUM LOAD 700 LBS. STEAM PER HOUR

Fig. 6—Example No. 4. The Boiler, at 100 Lb Pressure, Furnishes Steam for Heating and High Pressure Steam for Process Work

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0810 to .0850 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0810	.0820	.0830	.0840	.0850
100′	2	⅜	17.00	17.10	17.20	17.26	17.35
125′	2	½	33.30	33.50	33.70	33.90	34.10
150′	3	¾	76.30	76.80	77.30	77.75	78.20
	4	1	153.7	154.6	155.6	156.0	157.5
	5	1¼	338.0	340.0	342.0	344.0	346.0
	5	1½	525.0	528.5	532.0	535.0	538.0
	7	2	1064	1070	1077	1084	1090
	8	2½	1753	1765	1774	1785	1795
	10	3	3210	3230	3250	3270	3288
	12	3½	4800	4825	4855	4885	4912
	14	4	6778	6819	6861	6902	6949
	15	4½	9223	9278	9336	9391	9446
	17	5	12548	12623	12703	12778	12853
	20	6	20591	20714	20844	20967	21090
	24	7	30502	30684	30877	31059	31241
	27	8	42875	43131	43402	43658	43914
	30	9	57929.0	58275.0	58641.4	58987.5	59333.5
	34	10	78238.5	78705.8	79200.7	79668.0	80135.3
	40	12	125571.4	126321.5	127115.7	127865.7	128615.8
	47	14	162628.1	163599.6	164628.1	165599.6	166571.0
	53	16	234457.2	235857.7	237340.6	238741.1	240141.6
	60	18	322643.9	324571.1	326611.7	328539.0	330466.2
	67	20	428312.8	430871.2	433580.2	436138.6	438696.9
	81	24	696159.0	700317.4	704720.4	708878.7	713037.5

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0860 to .0900 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0860	.0870	.0880	.0890	.0900
100′	2	⅜	17.5	17.6	17.7	17.83	17.9
125′	2	½	34.3	34.5	34.7	34.9	35.1
150′	3	¾	78.6	79.1	79.5	80.0	80.45
	4	1	158.3	159.3	160.2	161	162
	5	1¼	348	350	352	354	356.3
	5	1½	541.5	544.5	547.5	550.5	554
	7	2	1096	1103	1108	1115	1122
	8	2½	1806	1814	1826	1837	1848
	10	3	3308	3325	3345	3365	3385
	12	3½	4940	4970	5000	5015	5028
	14	4	6983	7023	7064	7104	7145
	15	4½	9501	9557	9612	9667	9722
	17	5	12927	13002	13077	13152	13227
	20	6	21213	21336	21459	21582	21705
	24	7	31423	31605	31788	31970	32152
	27	8	44171	44427	44683	44939	45195
	30	9	59679.5	60025.5	60371.6	60717.6	61063.6
	34	10	80602.7	81070	81537.4	82004.7	82472.1
	40	12	129365.9	130116	130866	131616.1	132366.2
	47	14	167542.4	168513.8	169485.3	170456.7	171428.1
	53	16	241542	242942.5	244343	245743.5	247144
	60	18	332393.5	334320.7	336248	338175.2	340102.5
	67	20	441255.1	443814.2	446372.1	448930.2	451489.3
	81	24	717195.4	721353.8	725512.2	729670.5	733828.9

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0910 to .0950 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0910	.0920	.0930	.0940	.0950
100′	2	⅜	17.98	18.1	18.2	18.3	18.4
125′	2	½	35.3	35.45	35.7	35.9	36.1
150′	3	¾	80.9	81.33	81.75	82.2	82.62
	4	1	162.8	163.8	164.7	165.5	166.4
	5	1¼	358	360.2	362	364	366
	5	1½	556.8	560	563	566	569
	7	2	1127	1134	1140	1146	1153
	8	2½	1856	1868	1877	1888	1899
	10	3	3400	3422	3440	3458	3479
	12	3½	5085	5115	5140	5170	5190
	14	4	7183	7223	7262	7300	7340
	15	4½	9774	9829	9881	9932	9988
	17	5	13298	13373	13443	13514	13589
	20	6	21821	21944	22059	22175	22298
	24	7	32323	32506	32677	32849	33031
	27	8	45436	45692	45933	46174	46430
	30	9	61389.3	61735.3	62061	62386.7	62732.7
	34	10	82911.9	83379.2	83819.1	84259	84726.3
	40	12	133072.1	133822.2	134528.2	135234.1	135984.2
	47	14	172342.4	173313.8	174228.1	175142.4	176113.8
	53	16	248462.1	249862.6	251180.7	252498.8	253899.2
	60	18	341916.3	343843.6	345657.5	347471.3	349398.6
	67	20	453897.2	456455.6	458863.6	461271.5	463829.9
	81	24	737742.7	741901	745814.8	749728.5	753886.9

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0960 to .1000 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0960	.0970	.0980	.0990	.1000
100′	2	⅜	18.47	18.55	18.67	18.77	18.87
125′	2	½	36.22	36.45	36.6	36.8	37.0
150′	3	¾	83.1	83.5	83.95	84.35	84.8
	4	1	167.3	168.25	169	169.8	170.8
	5	1¼	368	370	371.8	373.5	375.5
	5	1½	572	575	578	581	584
	7	2	1158	1165	1170	1176	1182
	8	2½	1907	1918	1927	1937	1947
	10	3	3495	3512	3530	3548	3568
	12	3½	5220	5250	5275	5300	5330
	14	4	7378	7416	7454	7493	7531
	15	4½	10039	10091	10143	10195	10247
	17	5	13659	13730	13800	13871	13942
	20	6	22414	22530	22645	22761	22877
	24	7	33202	33374	33545	33717	33888
	27	8	46671	46912	47153	47394	47634
	30	9	63058.4	63384	63709.7	64035.4	64361.1
	34	10	85166.1	85606	86045.8	86485.7	86925.5
	40	12	136690.1	137396.1	138102	138808	139514
	47	14	177028.1	177942.4	178856.7	179770.9	180685.2
	53	16	255217.3	256535.4	257853.5	259171.6	260489.7
	60	18	351212.5	353026.3	354840.2	356654.1	358468
	67	20	466237.9	468645.8	471053.8	473461.7	475869.7
	81	24	757800.6	761714.4	765628.2	769541.9	773455.7

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .1250 to .2250 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.1250	.1500	.1750	.2000	.2250
125′	2	⅜	21.09	23.1	24.95	26.7	28.3
150′	2	½	41.35	45.3	49.0	52.3	55.5
	3	¾	94.8	103.8	112.1	119.85	127.15
	4	1	190.8	209.2	225.7	241.5	256.2
	5	1¼	420	460	496.5	531	563
	5	1½	652.5	715	772	825	875
	7	2	1322	1448	1564	1672	1773
	8	2½	2178	2386	2577	2756	2922
	10	3	3988	4370	4720	5045	5255
	12	3½	5960	6525	6880	7540	7985
	14	4	8419	9224	9962	10651	11296
	15	4½	11456	12551	13556	14492	15371
	17	5	15586	17076	18443	19718	20912
	20	6	25576	28021	30264	32355	34316
	24	7	37886	41508	44831	47928	50832
	27	8	53255	58347	63017	67371	71453
	30	9	71953.3	78833.1	85143.1	91025.5	96541.6
	34	10	97179.6	106471.4	114993.6	122938.4	130388.3
	40	12	155971.5	170884.8	184562.6	197313.9	209271
	47	14	201999.5	221313.7	239027.9	255542.2	271027.9
	53	16	291218	319062.9	344601.1	368409.3	390734.6
	60	18	400754.1	439072.3	474216.2	506979.4	537702
	67	20	532004.8	582872.6	629526.5	673020	713804.5
	81	24	864695.1	947373.1	1023202.1	1093894.3	1160183.5

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .2500 to .3500 Lb.

Velocity. Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.2500	.2750	.3000	.3250	.3500
	2	⅜	29.8	31.3	32.65	34.0	35.27
	2	½	58.5	61.4	64.0	66.7	69.2
	3	¾	134	140.5	146.6	152.7	158.3
	4	1	270	283	295.7	307.8	319.4
	5	1¼	594	623	650	677	702
	5	1½	923	969	1010	1052	1092
	7	2	1870	1962	2047	2130	2212
	8	2½	3080	3230	3370	3510	3680
	10	3	5570	5920	6175	6430	6675
	12	3½	8430	8840	9225	9600	9960
	14	4	11908	12490	13045	13573	14087
	15	4½	16203	16994	17749	18472	19172
	17	5	22046	23121	24149	25132	26084
	20	6	36175	37940	39626	41239	42802
	24	7	53587	56202	58699	61089	63404
	27	8	75325	79001	82511	85870	89124
	30	9	101772.7	106739.2	111481.8	116020.9	120417.5
	34	10	137453.5	144161.2	150566.5	156697	162634.9
	40	12	220610.3	231376.1	241656.6	251495.8	261026.2
	47	14	285713.6	299656.4	312970.6	325713.5	338056.3
	53	16	411906.6	432007.7	451202.5	469573.6	487368
	60	18	566837.5	594499.1	620913.7	646194.7	670682.1
	67	20	752482.1	789203.2	824268.9	857829.6	890336.8
	81	24	1223048.2	1282733	1339727	1394275	1447110.6

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .3750 to .4750 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.3750	.4000	.4250	.4500	.4750
	2	⅜	36.5	37.7	38.9	40.0	41.0
	2	½	71.6	74.0	76.3	78.5	80.7
	3	¾	164.1	169.5	174.8	179.7	184.7
	4	1	330.7	342.5	352.4	362.3	373.8
	5	1¼	727	751.5	775	796.8	818.5
	5	1½	1130	1167	1204	1238	1272
	7	2	2290	2365	2440	2508	2576
	8	2½	3775	3990	4020	4128	4245
	10	3	6910	7135	7365	7570	7770
	12	3½	10325	10660	11000	11325	11620
	14	4	14583	15064	15543	15976	16415
	15	4½	19843	20497	21149	21739	22335
	17	5	26997	27888	28774	29576	30386
	20	6	58046	59508	60962	62279	63610
	24	7	65622	67787	69941	71892	73864
	27	8	92243	95286	98314	101056	103828
	30	9	124630.9	128742.5	132833.8	136538.3	140283.5
	34	10	168325.5	173878.6	179404.3	184407.6	189465.9
	40	12	270159.4	279072.1	287940.6	295970.8	304089.3
	47	14	349884.8	361427.7	372913.3	383313.3	393827.6
	53	16	504420.9	521061.9	537620.6	552614	567772.1
	60	18	694149.2	717049.4	739836.3	760469.1	781328.8
	67	20	921489.6	951889.9	982139.7	1009530	1037221.3
	81	24	1497744.8	1547156	1596322.5	1640841.5	1685849.7

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .5000 to 1.0000 Lb.

Velocity F . Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot					
			.5000	.6000	.7000	.8000	.9000	1.0000
	2	⅜	42.2	46.2	49.85	53.35	56.6	59.7
	2	½	82.7	90.7	98.0	104.6	111	·11.7
	3	¾	189.4	207.6	224.3	239.7	254.3	268.16
	4	1	378.4	418.4	452	483	512.5	540
	5	1¼	839.5	920	994	1062	1126.5	1187
	5	1½	1305	1429	1544	1650	1751	1845
	7	2	2643	2896	3128	3344	3540	3738
	8	2½	4355	4775	5150	5501	5775	6159
	10	3	7980	8740	9440	10090	10700	11279
	12	3½	11920	13050	14000	15050	16000	16850
	14	4	16841	18449	19928	21302	22955	23817
	15	4½	22915	25102	27115	28985	30745	32407
	17	5	31177	34153	36891	39435	41829	44091
	20	6	51159	56042	60535	64710	68638	72349
	24	7	75782	83016	89672	95856	101675	107173
	27	8	106524	116693	126049	134741	142921	150649
	30	9	143927	157666.3	170306.5	182051.1	193103.6	203545.5
	34	10	194386.7	212942.9	230014.7	245876.8	260804.2	274907
	40	12	311987.1	341769.6	369169.4	394627.8	418586.1	441220.7
	47	14	404056.1	442627.5	478113.1	511084.4	542112.9	571427.2
	53	16	582518.4	638125.8	689284.6	736818.6	781551.7	823813.3
	60	18	801621.5	878144.6	948545.8	1013958.9	1075517.4	1133675
	67	20	1064160.2	1165745.3	1259203.6	1346040	1427759.6	1504964 3
	81	24	1729634.8	1894746.3	2046648.9	2187788.7	2320611	2446096

Pounds of Steam Carried in Pipe
for
Drops per Lineal Foot of .0660 to .0700 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Deop Per Lineal Foot				
			.0660	.0670	.0680	.0690	.0700
100′	2	⅜	15.33	15.45	15.55	15.67	15.77
125′	2	½	30.05	30.3	30.45	30.7	30.95
150′	3	¾	68.9	69.4	69.9	70.4	70.92
	4	1	138.76	139.8	140.8	141.8	142.85
	5	1¼	305	307.4	309.6	311.8	314
	5	1½	474.2	478	481.3	485	488.5
	7	2	961	968	974	982	989
	8	2½	1583	1594	1605	1617	1630
	10	3	[illegible]	2918	2940	2961	

Pounds of Steam Carried in Pipe
for
Drops per Lineal Foot of .0610 to .0650 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0610	.0620	.0630	.0640	.0650
	2	⅜	14.73	14.85	14.98	15.1	15.21
100′	2	½	28.9	29.15	29.4	29.6	29.8
125′	3	¾	66.2	66.8	67.3	67.85	68.35
150′	4	1	133.35	134.5	135.6	136.65	137.65
	5	1¼	293.2	295.7	298	300.5	302.7
	5	1½	455.8	459.8	463.3	467	470.5
	7	2	923	[illegible]	939	946	953
	8	2½	[illegible]			[illegible]	

Above—A 2-In. Pipe Will Carry 982 Lb of Steam Per Hour With a Drop Per Lineal Foot of .0690 Lb.

Below—A 1¼-In. Pipe Will Carry 293.2 Lb of Steam Per Hour With a Drop Per Lineal Foot of .0610 Lb.

ing the actual load by 2.40. This may at first seem rather misleading but nevertheless is usually the easiest and most practical method of conversion and is correct.

Piece Mark	Load Lbs.	Corrected Load, Lbs.	Length Ft.	Ells	Equiv. Lgth.	Size In.	Drop per Ft.	Drop in Piece
Z	2345	977	260	7	309	2	.069	21.32
X	700	292	70	6	100	1¼	.061	6.10
		Totals	330	13				27.42

Length of Longest Run

After totalling the actual length of the longest run and finding this to be 330 ft. and 13 ells, it will be safe to say that the maximum equivalent length will not exceed 400 ft. and therefore this figure may be used to find the preliminary approximate drop in pressure per foot which will be permissible. Dividing 40 lb. allowable total drop by 400 ft. gives us an average of .100 lb. per ft.

The engineer experienced with this method of calculation would readily pick out a 2-in. pipe for this load but just for purpose of illustration the actual drop in the first pipe or main line *Z* will be figured on the basis of three sizes of pipe.

Corrected Load Lb.	Length Ft.	Ells	Equiv. Lgth.	Size Inches	Drop per Ft.	Total Drop
977	260	7	295	1½	.280	82.60
977	260	7	309	2	.069	21.32
977	260	7	316	2½	.025	7.90

The foregoing calculations for three sizes illustrate that the 1½-in. size is out of the question but 2-in. is permissible. Of course the engineer must always consider the possibilities for future loads which may be added but in this example this has not been anticipated and consequently 2-in. will be entered into our tabulation as the size for the main pipe from boiler header to tee *Y*. In calculating the size of line *X* supplying the dryers it will be found that a practical yet economical size for this will be 1¼ in. The total drop from boiler header to farthest point will be therefore (under maximum load conditions) 27.42 lb.

Second step: The second and remaining step in this example is to calculate the size of line *W* which supplies steam from "common tee" *Y* through the pressure reducing valve to the heating system. Again three sizes of pipe will be figured for purposes of illustration.

Load Lb.	Corrected Load	Length Ft.	Ells	Equiv. Lgth.	Size Inches	Drop per Ft.	Total Drop
1645	685	20	2	30	1¼	.330	9.90
1645	685	20	2	30	1½	.136	4.08
1645	685	20	2	34	2	.0335	1.14

In the figuring of pipe sizes for high pressure piping in general the amount of steam utilized is usually "metered" through pressure reducing valves, control valves (either automatic or hand type), turbine nozzles or other devices which to a large degree minimize the value of equalizing the pressure drop to the various terminals of supply. Excepting in the case of supplying steam to certain high pressure heating units, all of which return into a common discharge, there is therefore little to be gained by attempting to equalize the pressure drop in various branches of high pressure steam lines. Therefore these pressure drop calculations are useful only as they indicate the minimum size permissible or the practical sizes.

In this example, the 1¼-in. size is clearly permissible and could well be used for the by-pass connection around the pressure reducing valve as well as the supply to it. Of course, in this particular example, the sizes are so small anyway that there is little economy in using 1¼-in. versus 1½-in. or 2-in. and therefore probably the most practical answer is to use 1½-in. which would fit in with the opening of a standard 1½ × 3-in. pressure reducing valve. However, the principle of these calculations would be identical for a system many times as large

Pounds of Steam Carried in Pipe
for
Drops per Lineal Foot of .0330 to .0350 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0330	.0335	.0340	.0345	.0350
	2	⅜	10.83	10.92	11.0	11.08	11.16
75′	2	½	21.25	21.4	21.6	21.7	21.9
100′	3	¾	48.7	49.08	49.45	49.8	50.12
125′	4	1	98.1	98.85	99.6	100.3	101
150′	5	1¼	215.7	217.3	219	220.5	222
	5	1½	335.2	338	340.3	342.7	345.2
	7	2	679	684	689.7	694.5	699.5
	8	2½	1118	[illegible]	1135	1143	1152

A 2-In. Pipe Will Carry 684 Lb of Steam Per Hour With a Drop Per Lineal Foot of .0335 Lb.

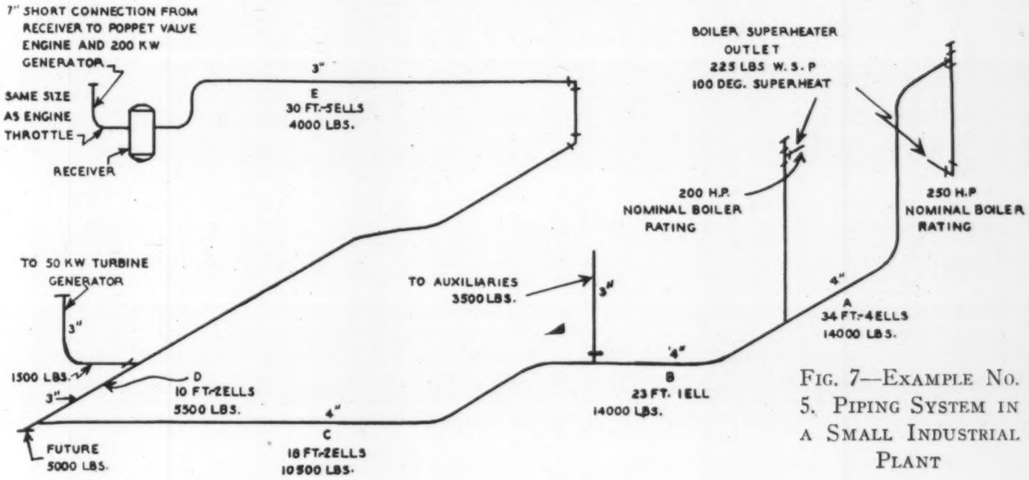

7" SHORT CONNECTION FROM RECEIVER TO POPPET VALVE ENGINE AND 200 KW GENERATOR
SAME SIZE AS ENGINE THROTTLE
RECEIVER
3"
E
30 FT.-5ELLS
4000 LBS.
BOILER SUPERHEATER OUTLET
225 LBS W. S. P
100 DEG. SUPERHEAT
200 H.P. NOMINAL BOILER RATING
250 H.P NOMINAL BOILER RATING
TO 50 KW TURBINE GENERATOR
3"
TO AUXILIARIES
3500 LBS.
3"
4"
A
34 FT.-4ELLS
14000 LBS.
4"
B
23 FT. 1ELL
14000 LBS.
1500 LBS.
3"
D
10 FT.-2ELLS
5500 LBS.
4"
C
18 FT.-2ELLS
10500 LBS.
FUTURE
5000 LBS.

FIG. 7—EXAMPLE NO. 5. PIPING SYSTEM IN A SMALL INDUSTRIAL PLANT

as this one and there the economy in first cost, especially where several valves are required, the saving in heat loss, etc., all would point to the advisability of using the minimum permissible size of connection.

EXAMPLE NO. 5—
POWER PLANT PIPING

As already explained in connection with the preceding example, equalization of pressures at the various terminals of the branch lines of a high pressure steam piping system is only of interest in certain high pressure *heating* or allied applications. In power plant piping equalization is of no interest. The three factors that govern the choice of pipe sizes for this class of piping are:

a. Loss in pressure due to friction
b. First cost, maintenance cost, and depreciation
c. Heat loss by radiation

Regardless of the size of plant, the steam pressure or the steam temperatures involved, these tables with the conversion factors may be used for determining the practical and yet economical sizes of pipe. For important piping lines of considerable length and size, the method (used in Example 4) of figuring three sizes of pipe will usually permit the engineer designing the piping system to pick readily a size which will thorougly satisfy him and will leave no question in his mind but that he has decided upon exactly the right size.

It is true that often future expansion has not been allowed for in piping where it would have been advantageous to have done so. On the other hand, probably in many times the number of cases where the foregoing condition has developed, pipes many sizes larger than could ever be used have been installed with consequent waste due to excessive first cost, excessive heat loss and maintenance.

Fig. 7 is a diagram of a piping system installed about 10 years ago in a small industrial power plant. The

Pounds of Steam Carried in Pipe
for
Drops per Lineal Foot of .0305 to .0325 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0305	.0310	.0315	.0320	.0325
	2	⅜	10.42	10.5	10.59	10.68	10.75
	2	½	20.42	20.6	20.75		21.08
	8	2½		1084	1093		1110
	10	3	1970	1985	2001	2020	2032
	12	3½	2943	2965	2990	3013	3035
	14	4	4158	4191	4227	4261	4291
	15	4½	5658.3	5703.7	5752.3	5797.7	5839.8
	17	5	7698	7760	7826	7888	7945
	20	6	12632	12734	12842	12943	13037
	24	7	18712	18862	19023	19173	

Pounds of Steam Carried in Pipe
for
Drops per Lineal Foot of .0182 to .0190 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0182	.0184	.0186	.0188	.0190
50′	2	⅜	8.05	8.09	8.14	8.18	8.225
	8	2½	830	835	840		
	10	3	1522	1530	1536	1546	1555
	12	3½	2275	2285	2298	2310	2322
	14	4	3213	3230	3245	3262	3280
	15	4½	4371	4395	4419	4443	4467
	17	5					

ABOVE—A 4-IN. PIPE WILL CARRY 4,261 LB OF STEAM PER HOUR WITH A DROP PER LINEAL FOOT OF .0320 LB.

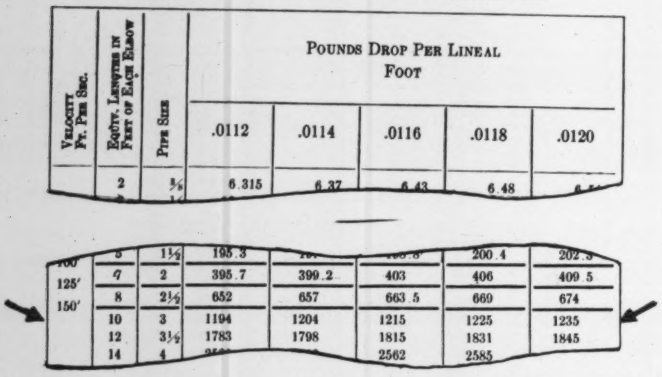
BELOW—A 4-IN. PIPE WILL CARRY 3,213 LB OF STEAM PER HOUR WITH A DROP PER LINEAL FOOT OF .0182 LB

Pounds of Steam Carried in Pipe
for
Drops per Lineal Foot of .0230 to .0250 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0230	.0235	.0240	.0245	.0250
50′	2	⅜	9.05	9.15	9.24	9.34	9.435
75′	2	½	17.73	17.93	18.13	18.32	18.5
	3	¾	40.65	41.1	41.54	41.96	42.4
100′	4	1	81.9	82.8	83.7	84.5	85.4
125′	5	1¼	180	182	184	185.8	187.7
150′	5	1½	280	283	286	289	292
	7	2	567	573.5	579	585	591
	8	2½	934	944	954	964	974
	10	3	1710	1730	1748	1765	1783
	12	3½	2555	2579	2610	2635	2662
						3727.4	

A 3-IN. PIPE WILL CARRY 1.710 LB OF STEAM PER HOUR WITH A DROP PER LINEAL FOOT OF .0230 LB.

Pounds of Steam Carried in Pipe
for
Drops per Lineal Foot of .0112 to .0120 Lb.



Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0112	.0114	.0116	.0118	.0120
	2	⅜	6.315	6.37	6.43	6.48	[illegible]
100'	5	1½	195.3	[illegible]	[illegible]	200.4	[illegible]
125'	7	2	395.7	399.2	403	406	409.5
150'	8	2½	652	657	663.5	669	674
	10	3	1194	1204	1215	1225	1235
	12	3½	1783	1798	1815	1831	1845
	14	4	[illegible]	[illegible]	2562	2585	[illegible]

A 3-In. Pipe Will Carry 1,220 Lb of Steam Per Hour With a Drop Per Lineal Foot of .0117 Lb.

pipe sizes were figured on the basis of this data. Without going into too much repetitive detail of explanation, the following tabulation gives the calculations for the main run to the farthest unit on this high pressure distribution system.

Piece Mark	Load Lbs.	*Corrected Load	Length Ft.	Ells	Equiv. Lgth.	Size In.	Drop per Ft.	Drop in Piece	Total Drop
A	14000	4268	34	4	90	4	.032	2.88	
B	14000	4268	23	1	37	4	.032	1.184	4.064
C	10500	3201	18	2	46	4	.0181	.8326	4.897
D	5500	1676	10	2	30	3	.023	.690	5.587
E	4000	1220	30	5	80	3	.0117	.936	6.523

This actual installation was in operation for several years before the "future load" noted on the plans was added in the form of an additional poppet valve engine and generator. The system actually is operating with somewhat less total drop in pressure than the theoretical, because it is rarely if ever that the entire maximum load is carried by the one boiler, whereas these calculations are based on that possibility when figuring the total maximum load on piece *A*.

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*In this case the corrected load has been found with the conversion factor for 200 lb. pressure and 100 deg. superheat, which is 3.28.

Chemical Piping Oxy-acetylene Welded from Sheet Aluminum

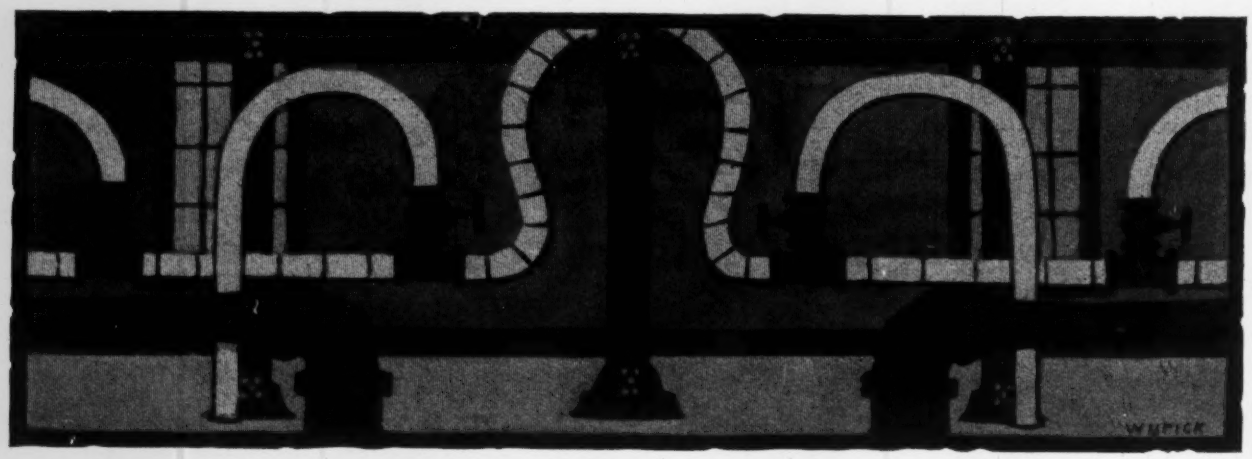
Sheet aluminum is the commercially pure metal, while cast aluminum is alloyed with an appreciable percentage of other metals. The rolled metal never contains more than 2 per cent of other metallic ingredients and frequently less than 1 per cent. The chief impurities in aluminum sheet are iron and silicon but these do not appreciably affect its welding qualities. The high rate of thermal conductivity and very low fusion point of aluminum are characteristics which must be studied by the welder. The aluminum oxide film has a higher melting point than the aluminum and therefore without proper fluxing this oxide film holds the molten aluminum until either the film melts or the molten aluminum ruptures the film, making it difficult to tell when the melting point has been reached.

This makes it necessary for the operator to develop a technique somewhat different from that employed in welding steel. A good grade of aluminum flux should be used to remove the oxide which forms readily on the surface of molten aluminum. True fusion takes place only after the flux has been applied, removing the oxide.

Aluminum sheet should be prepared and set up for welding in about the same way as sheet steel except that its higher heat conductivity makes necessary a slightly greater spacing between the edges. For light sheet aluminum a flange weld is probably the most satisfactory. Sheet 1/16 to ⅛ in. in thickness should be joined with a butt weld. It is not, as a rule, necessary to bevel the edges. It is advisable to support the sheet during welding because aluminum is rather weak when hot. Many operators find it advantageous to support the under side with pieces of steel sheet or plate.

Making Elbows

On one eastern job which called for making chemical pipe from sheet aluminum, the elbows were made by heating the parts, bending them to shape, and quenching them instead of hammering the elbows into shape.—*Oxy-acetylene Tips.*



The Maintenance of Air Conditioning Equipment in Industrial Plants

By William A. Hanley*

The engineer will find suggestions for a complete inspection and maintenance routine for air conditioning equipment in this article. Mr. Hanley has had wide experience as director of engineering for a large manufacturing concern; his paper on this important, though sometimes slighted, subject cannot help but be of value. It is the type of article you will find posted for ready reference in the plant.

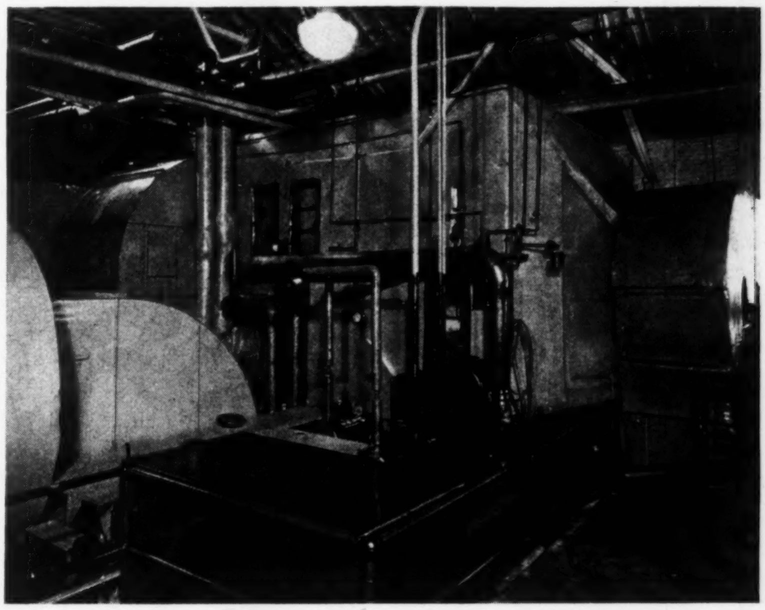
THE problem of maintenance of air conditioning equipment offers no great difficulties and in the main is relatively simple. Since, however, there are moving parts in every air conditioning plant, that one of the functions of the plant is to remove dirt, and that conditions are often favorable to oxidation or the formation of rust due to the presence of water and air, proper maintenance is a necessary function of good operation.

Advantages of Proper Maintenance

The properly maintained plant will last longer, cost less to operate, and run more regularly than one which is neglected. Every air conditioning plant should be in charge of some individual who will look after the routine operation of it and who is familiar with its functions. This individual need not be an engineer or an expert mechanic necessarily, but should be a person who will exercise ordinary judgment and who knows how to perform simple mechanical operations. If the air conditioning plant is started and stopped, and otherwise looked after by a group of people, it will be a case of everybody's business is nobody's business and the plant will soon be neglected.

Daily Inspection Routine

Every day the plant is in operation the individual in charge should check the pressure on the spray heads, examine the dirt screens and should see if predetermined temperatures and pressures are being maintained. This information should be recorded, because where the information is not recorded, pressures may drop one pound per week and the drop not be perceptible to the ordinary individual or be considered negligible. Where recording instruments are used, this information is automatically recorded as regards temperatures and pressures; and where rotary screens are supplied they will largely take care



Showing Air Conditioning Equipment in an Industrial Plant Cleaned and Painted and in Proper Condition for Efficient Operation of themselves. In the smaller installations, however, where there are no recording instruments, we cannot advocate too strongly the matter of some individual recording daily the pressures and temperatures of the air conditioning equipment. Any variation which tends to creep into the operation can be detected quickly and corrected.

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* Eli Lilly & Company, Indianapolis, Ind.

Condition of Spray Heads

The efficiency of any air washing unit is largely a function of the condition of the spray heads. Depending upon what is passing into the washer, the spray heads should be cleaned daily, weekly or monthly. With larger nozzle orifices on spray heads the problem is somewhat simplified, but as all the pipe lines to the spray heads have a tendency to collect foreign matter, they should be thoroughly washed out from time to time.

Cleaning Intake Ducts

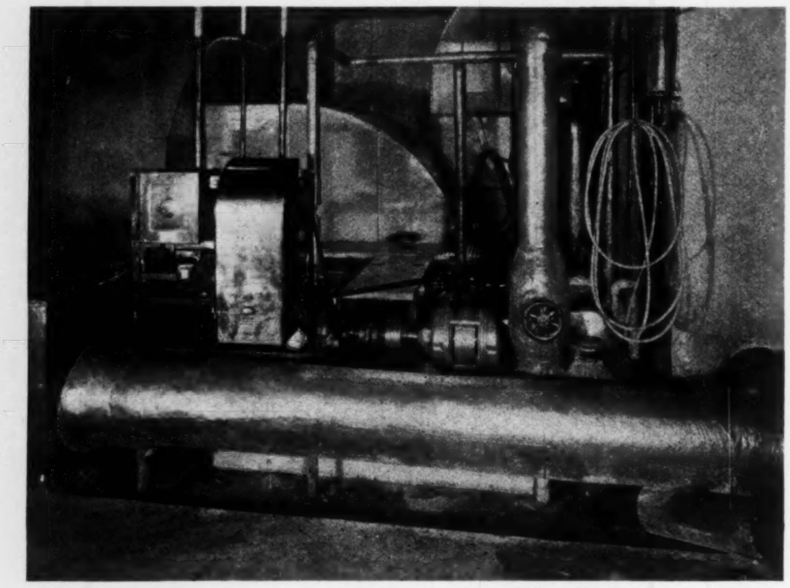
There is in every air conditioning plant an intake or set of return air ducts which are bound to accumulate dirt. This is the dirt which settles out before it reaches the washer. In some plants it is necessary to clean these ducts or this intake monthly, but in most instances a quarterly or semi-annual cleaning will keep conditions satisfactory.

Annual Overhauling

At least once in every year the air conditioning equipment should be shut down completely for several days. During that time the entire interior of the washer, unless it is made of copper, should be painted; all places which are tending to rust should be cleaned, and the entire plant gone over and put in good condition. There will always be found some pipes which are dripping and some insulation which is loose and this should all be fixed and everything cleaned up and made attractive. Before any painting is done on the interior of the washer, it should be thoroughly dried out. This of course holds true for not only the interior of the washer, but any other surface which is to be painted.

Take Care of Bearings

It seems hardly necessary to say that the bearings of electric motors and centrifugal pumps should be taken care of at regular intervals. The ordinary sleeve bearings equipped with ring oilers will run twenty-four hours per day for thirty days under most operating conditions with one filling. Ball bearings and roller bearings will run longer; but at the end of every year when the annual clean up is made, the bearings should be thoroughly washed out and refilled with fresh oil or grease.



The Centrifugal Pumps Are Placed in Pits in This Plant to Take Care of the Drip. Note the Walkways Over the Pipe Covering, with Valve Wheels Extended Through the Walkways

Replace Nozzle Orifices

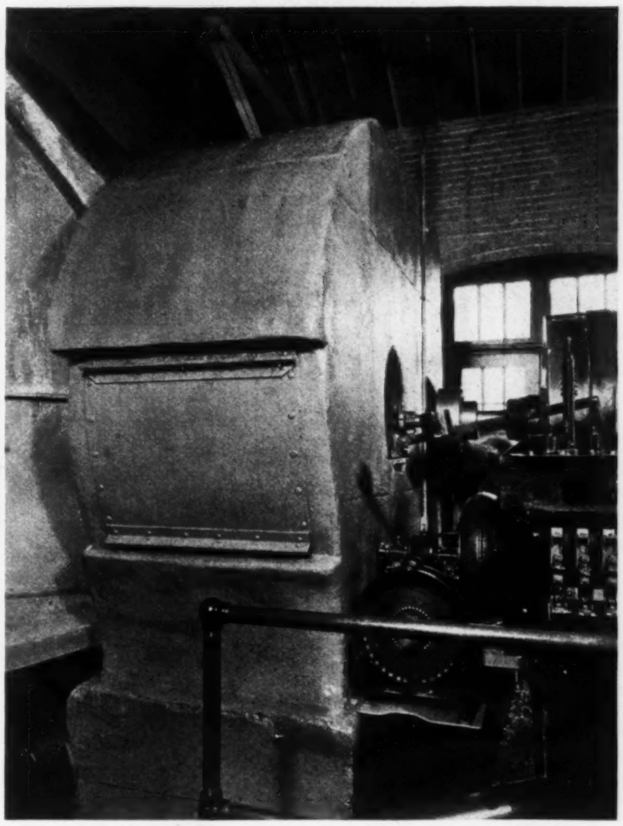
About once every five years under ordinary conditions nozzle orifices must be replaced. This is a comparatively simple job, but with the enlargement of these orifices the volume of water circulated increases and the pressure on the nozzles is reduced. This reduction in pressure on the nozzles cuts down the atomization and it will be found that the difference in temperature between the air leaving the washer, and the water in the washer, is increased, which means that the heat transfer from air to water is becoming less efficient. The replacement of nozzle orifices will correct this condition.

Care of Centrifugal Pump

The centrifugal pump is a rugged piece of machinery. The tendency of the amateur is to pack the centrifugal pump, so as to avoid dripping. This, however, is a false economy and it is much better to have a slight drip at the stuffing boxes than to have the pump packed so tight that it will be necessary to replace the pump shaft every year or two. Provision should be made to take care of the dripping which comes from these stuffing boxes when the installation is made, and then the pump shaft will last as long as the centrifugal pump itself, which, with ordinary care, is usually between fifteen and twenty years.

Eliminators and Fan Runner

Each year when the plant is shut down for the annual overhauling and the interior of the washer is cleaned and painted, particular attention should be paid to the eliminators and the fan runner. Our experience has been that this part of the equipment tends to erode faster than any other part of the plant. When the washer is installed, doors should be provided so that this runner is easily accessible and so that the eliminators are easily removable for cleaning and painting or replacement. Galvanized iron eliminators will ordinarily last from eight to ten years and the fan runner from twelve to fifteen years.



Note the Door Installed in the Fan Runner Housing for Cleaning and Ease in Maintenance

Process Work Influences Stock of Spare Parts

For important process work, dependent on the output of an air conditioning plant, certain precautions are justified which would not be reasonable under ordinary conditions. A process may be started involving some very valuable materials, where a failure of the air conditioning plant will mean a loss of several thousand dollars. In such instances it is very wise to have on hand spare parts to replace anything which may be broken and to do a much better job of inspection and maintenance.

Diaphragms and Gaskets

The matter of diaphragms and gaskets alone has caused much trouble in the past. Where important process work is being done we have found it profitable to annually replace all non-metallic gaskets and diaphragms and all metallic gaskets and diaphragms every three to five years. In churches, theaters, department stores and other public buildings, where a shutdown is not expensive, these precautions may not be justified; but all thermostats, no matter for what purpose they are used, should be thoroughly cleaned at least once a year. At that time the gaskets and diaphragms on these thermostats can be tested and any faulty ones replaced.

Air Conditioning Equipment Important

Air conditioning equipment has been a great help to industry and has become almost a necessity in public buildings, theaters, etc. A little intelligent management on the part of the owners in checking up the operator or maintenance man will be found to give splendid results. The man doing the operating or maintaining should be encouraged to keep everything clean and attractive and should be supplied with plenty of spare parts, paint, tools, etc. Where such equipment is provided the air conditioning plant will run regularly and be a source of satisfaction.

Mechanical Equipment of the $3,000,000 Scottish Rite Cathedral

THE SCOTTISH Rite Cathedral of Indianapolis, Ind., is the result of efforts on the part of George F. Schreiber, the architect, and Bevington-Williams, Inc., consulting engineers, both of Indianapolis, to fulfill the requirement for a structure which would attain a high plane in beauty, permanency and symbolism, as well as in the service of its mechanical equipment.

This structure, from which every engineer wholly concerned with utility could not withhold his admiration, occupies a ground space of 380 by 180 feet, has a cubic capacity of 3,418,940 cubic feet, consists of five stories above and two stories below grade, has a tower 214 feet high enclosing a carillon of bells and cost $2,850,000 exclusive of real estate. It is composed of lounge rooms, library, banquet hall, grand salon, auditorium, cafeteria, toilets and showers, billiard and card rooms and numerous dressing rooms, offices, conference and committee rooms.

As an indication of its permanency, the building is of steel and reinforced concrete skeleton construction. The entire exterior is of Indiana limestone with brick backing. As an example of the symbolism that is evident in every part of the building, the number "33" which represents the highest degree of Scottish Rite Masonry, is memorialized throughout the construction.

The performance of the mechanical equipment was given a capacity test at the time of the dedication of the building last Fall, when 3500 persons were seated at one time in the banquet hall, 3000 persons were entertained in the grand salon and official dedication ceremonies were performed before a large audience of members and notables from this and foreign countries in the auditorium, at which time all parts of the equipment performed satisfactorily.

Upon entering the building at the main entrance, the heating engineer will observe first the bronze grilles of special design concealing indirect heaters. There are 1200 grilles in the building, many of them designed especially for this job. The placing of these grilles in the paneling was a difficult task. Each supply grille is equipped with lock louvre damper which is set by means of a key through the face of the grille, the dampers moving upward to close. All vent grilles are also provided with lock louvre dampers which move downward to close. In some instances, grilles are located in window stools.

Some of the Engineering Problems

The first consideration in the design of the building was not that it house mechanical equipment but that in its beauty, its permanency and its symbolism it should typify faithfully the principles of the Scottish Rite Order. The most important engineering problem here was to provide not only suitable air, water and food supplies, elimination of smoke at banquets, elimination of odors, disposal of waste and absorption of noises, but to provide these services for a large number of people at short notice and at a reasonable operating expense. The membership of 11,000 persons requires that provision be made for a total occupancy of 7,000. For this reason, all equipment was designed for this peak load, which occurs at the time of convocations. How these objectives were achieved, how great flexibility is maintained and how the engineers solved the problem of placing the mechanical equipment, is the subject of this article.

Fuel Burning and Steam Generating Equipment

The steam generating plant is an excellent example of modern design. Two boilers, each rated at 219 hp., are installed for operating at a pressure of 25 lb. They are of the cross-drum box-header type and are fired by single retort, under-feed stokers designed for burning bituminous coal. One hundred and seventy-five per cent of rating is developed on each of these boilers when burning bituminous coal of 11,750 Btu. The cheapest grade of Indiana coal is used. The heating system is designed to provide suitable temperature in the building under ordinary conditions by the use of one boiler.

The Stokers

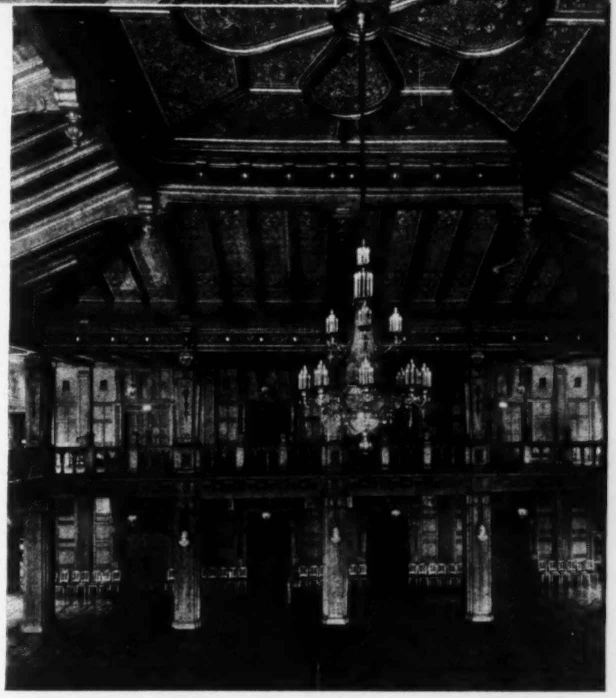
The stokers are driven by a 5-hp. motor through a variable speed device which is connected to an automatic regulator. The regulator is also connected to the drive of the forced draft fan which delivers air to the stokers. By this arrangement of control, both the stoker speed and fan speed are regulated automatically by variations in the steam pressure.

The Stoker Variable Speed Drive

Variable speed drives as used on these stokers also have many applications in regulating the speed of steam turbines, powdered coal feeders, etc., where it is necessary to change speed while the machinery is in motion. In this installation the drives are interposed between the motor and the stokers. The operating principle lies in its two parallel shafts with two metal discs on each shaft with their convex faces facing each other. A belt runs from one shaft to the other shaft between the discs. As the discs are moved farther apart, the belt sinks between them, closer to the shaft, decreasing the circumference of the drive and thus lessening the speed. In the same manner, when the discs are moved closer together, the speed is increased. One shaft is connected to the motor. The discs are moved closer together or farther apart by a steam pressure regulator so that the steam pressure regulates the speed at which the stoker feeds coal into the boiler. Spray pipes 1½ inches in diameter are installed in the ash pits of the stokers.

The Stack and Breeching

A suspended steel stack, 5 feet in diameter, extends 150 feet above the grates and is covered 2 inches thick



Above — The Carrilon Tower Showing the Effect of Flood-Lighting

Right — The Main Lobby Showing the Grilles for Ceiling Ventilation

Below — The Grand Salon Showing Acoustical Treatment of Ceiling

with 85 per cent magnesia throughout its entire length. It is sealed in a masonry shaft reaching from the boiler room to the top of the stack. The breeching is also covered with 85 per cent magnesia, 2 inches thick.

The Piping

There are four outlets from the boiler header, two 4 inches, one 6 inches and one 8 inches. The steam and return piping is concealed in the two main tunnels beneath the floor of the banquet hall.

One of the 4-in. mains supplies the direct radiation of the banquet hall and main floor. The other 4-in. main runs above the ceiling of the kitchen to supply the kitchen equipment with steam and the hot water heaters with steam. The 8-in. main runs above the ceiling of the kitchen to the pent house on the roof where it branches into two 5-in. mains, one to supply steam for the auditorium and the other to supply steam for the grand salon, with branches to the fan rooms. The 6-in. main supplies the indirect radiation for the banquet hall and main floor. All mains are dripped to the vacuum return. All steam mains and branches are covered one inch thick with 85 per cent magnesia.

The Heating

For the heating of this structure 10,712 square feet of cast iron indirect radiation and direct radiation, making a total of 80,000 square feet of equivalent direct radiation, were used, distributed as follows:

Indirect Radiation		
Banquet hall, 4 stacks deep, 2 tiers high	2,592 sq. ft.	
Grand salon, 4 stacks deep, 2 tiers high	2,688 sq. ft.	
Auditorium, 4 stacks deep, 2 tiers high	3,072 sq. ft.	
Main floor, 4 stacks deep, 2 tiers high	2,360 sq. ft.	
Other rooms, offices, etc.		E. D. R.
	10,712 sq. ft.	80,000 sq. ft.

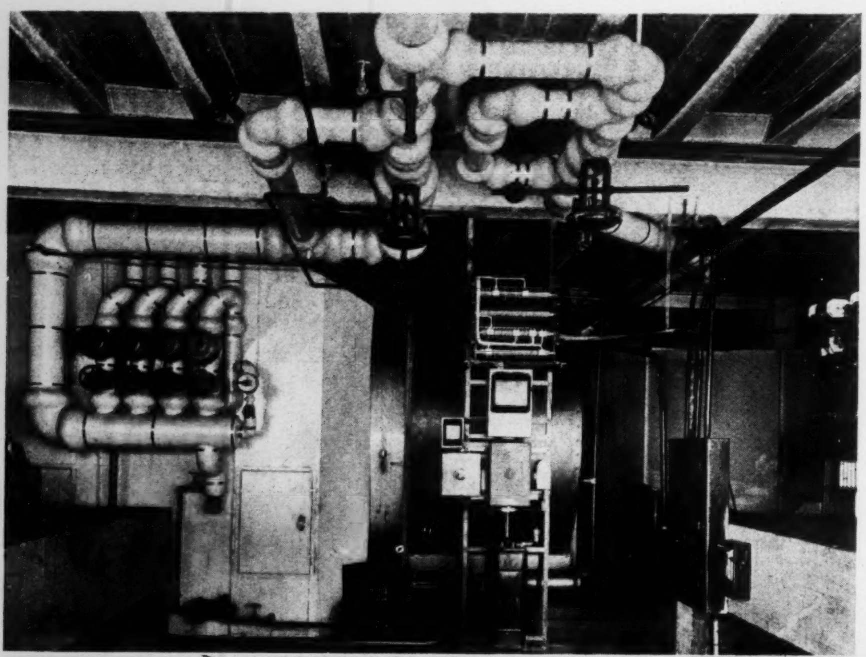
Most of the direct radiation is recessed in walls behind specially designed cast bronze grilles. Inlet and outlet grilles are also installed in the window stools. In the banquet hall a metal shield extends into the room above the recess spaces in order to catch the dust from the upward flow of air for the protection of the walls.

In the grand salon and auditorium, radiators are fed by downfeed risers concealed in the walls from the distribution piping in the attic. Radiators on the main floor are connected with a steam main and return piping in the main air tunnel which also serves the radiators located in the banquet hall.

Air Supply and Exhaust

The air supply is divided into four sections, one for the banquet hall, one for the grand salon, one for the auditorium and one for the main floor. Each is an individual system and the banquet hall and main floor are supplied from a common air intake in the fresh air room, drawing air through heaters and air filters in the rear of the boiler room at the northeast corner of the building. The units for the grand salon and auditorium each draw the air supply through louvred openings in each pent house. In order to save space in the building, the main supply and exhaust tunnels, 4 ft., 6 in. wide by 5 ft., 5 in. high, were made of concrete and run parallel to each other beneath the banquet room floor. In the remainder of the duct system 109 tons of sheet metal were used.

The banquet hall air supply is from the main concrete ducts beneath the floor, with sheet metal risers to grilles located approximately 8 feet above the main floor and 8 feet above the mezzanine floor. Lateral ducts connect trunk lines with exhaust ducts in the ceiling where they drop down at the north end of the hall to the main exhaust ducts.



A Section of the Fan Room for the Auditorium, Showing High and Low Pressure Steam Piping and Connections to Heaters and Humidifier

When 3,500 persons crowd into the banquet hall, ninety per cent of them smoking, the elimination of smoke was a problem. For the removal of smoke and to keep the air pure, 100,000 c.f.m. are supplied and exhausted for the peak load. Under ordinary conditions 25,000 c.f.m. are required. Under these minimum conditions, one of the fans is used with variable speed control and discharges into an auxiliary tunnel which is controlled by automatic dampers which close the main tunnel and open the smaller tunnel as required.

The foul air is exhausted from the banquet hall through grilles located at the main floor line, mezzanine floor line, and at the ceiling, the latter acting as smoke skimmers.

The Boiler Room, Showing Two 250 Horsepower Water Tube Boilers and Electric Driven Underfeed Stokers

The supply for the grand salon is taken through grilles in a pent house on the roof, discharging into trunk ducts and leading to grilles approximately 8 feet above the main and balcony floors. The foul air is removed through grilles at the main floor line and ceiling over the balcony which leads to the exhaust ducts in the attic.

The auditorium gets its air supply from a pent house discharging into trunk ducts which extend the length of the room to grilles at the floor line of the first and second balconies. The supply ducts extend to the stage, where two large warm air outlets are located. Foul air is exhausted at the floor line of the main floor and in the side walls near the ceiling connected with ducts which run to the exhaust fan in the pent house.

The main floor is supplied through ducts above the ceiling of the mezzanine floor and through riser ducts which lead to grilles approximately 8 feet above the main floor. The exhaust from floor line and side walls leads to riser ducts above the ceiling and to the exhaust chamber in the basement. The ducts for the kitchens run above the ceilings and connect with riser ducts connecting to the kitchen exhaust fan. The cafeteria can be ventilated from the banquet hall by means of a supply tunnel running from the plenum tunnel. The billiard and card room supply and exhaust ducts are also carried on the kitchen ceiling.

List of Fans Used

	Supply	Exhaust
Banquet Hall ..	2—50,000 c.f.m. 15 hp.	2—40,000 c.f.m. 10 hp.
Grand Salon....	1—40,000 c.f.m. 20 hp.	33,400 c.f.m. 10 hp.
Auditorium	1—41,500 c.f.m. 20 hp.	33,400 c.f.m. 10 hp.
Main Floor.....	1—35,000 c.f.m. 10 hp.	28,000 c.f.m. 10 hp.
Kitchens		3,600 c.f.m. 5 hp.
Toilet Rooms....		3,600 c.f.m. 5 hp.
Cafeteria		1,620 c.f.m. ½ hp.
		1,620 c.f.m. ½ hp.
	Constant speed	4,112 c.f.m. 1 hp.
		2,232 c.f.m. 6 hp.

Control of Temperature and Humidity

The direct radiation, indirect radiation and humidifying units are under automatic control by means of thermostats, diaphragm valves, damper motors and hygrostats actuated by compressed air at 15 pounds pressure provided by an automatic electric air compressor which maintains a constant pressure of 30 pounds per square inch on an air storage tank from which air is reduced to 15 pounds for the control system. Galvanized iron pipe lines run from the air storage tank to each thermostat and hygrostat while return lines run from each of these to the various diaphragm valves or motor dampers which they control and connect with thermostats and return valves with short flexible connections.

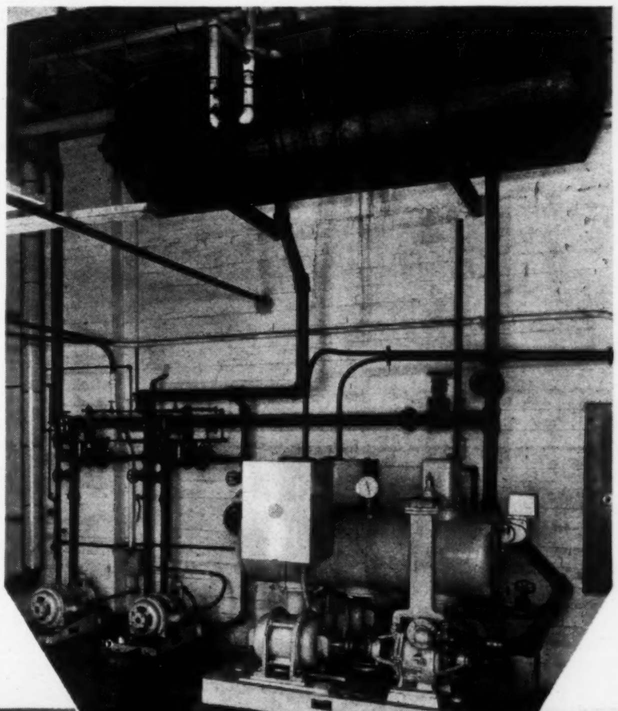
The cycle of operation as applied to the control of the direct radiators is as follows:

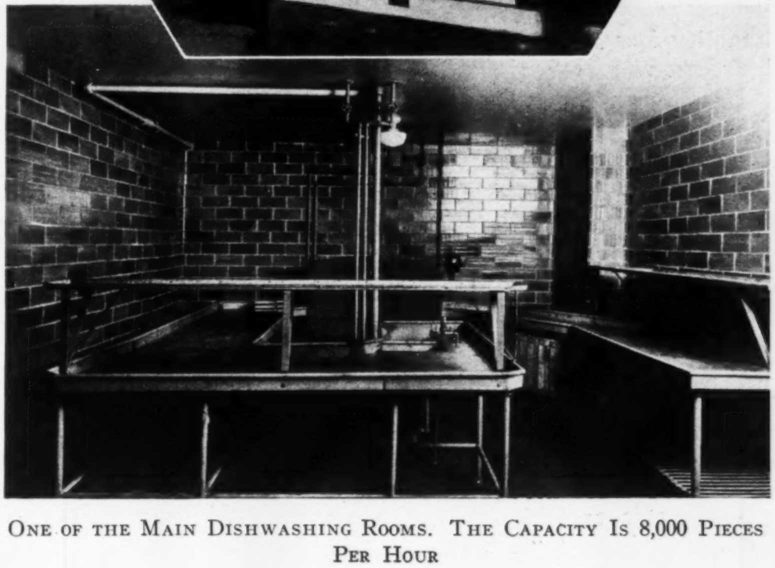
In the morning when the rooms are cool, the air supply valve in the thermostat is closed and the thermostat exhaust valve is open, thus relieving all air pressure from the diaphragm valve top and allowing a spring which is enclosed in the top to open the radiator valve. As the room temperature approaches the point for which the thermostat is set, the thermostat exhaust valve closes and the supply valve opens to pass air through the thermostat to the diaphragm valve top which starts to close the radiator valve. This double valve action of the thermostat, which is controlled by a vapor disc expanding and contracting in accordance with temperature changes, causes the thermostat to act as a reducing valve to admit just enough air to the diaphragm valve top or exhaust it to atmosphere to hold the radiator valve in whatever position necessary to admit enough steam to the radiators to maintain the desired room temperature. The pressure in the valve top may, therefore, vary from 0 to 15 pounds pressure, and the valve may be held in any position from fully open to fully closed.

For control of the various indirect heating units, duct-type thermostats located in the sheet metal ducts operate diaphragm valves which are placed in the steam supply lines to each section of heater coils. These coil valves are equipped with differential springs so that a drop in temperature of the incoming air will cause the heater coil nearest the fresh air intake to open first, the second one next, and so on, each closing in reverse order.

In connection with the duct thermostats, pilot thermostats are placed in the rooms heated by the various indirect heater units. These pilot thermostats control the supply of compressed air to their respective duct thermostats which in turn control the heater coils. This produces an action as follows:

Electric Driven Vacuum and Boiler Feed Pumps



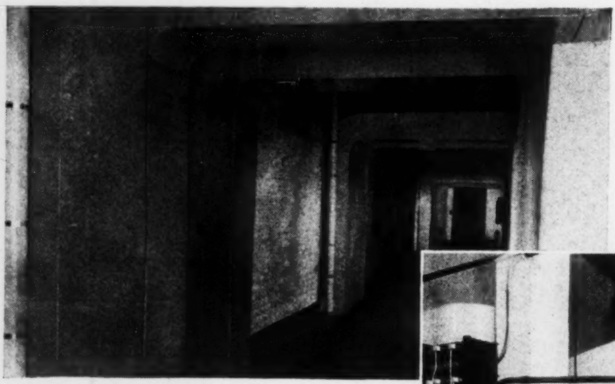


One of the Main Dishwashing Rooms. The Capacity Is 8,000 Pieces Per Hour

When the room is up to the temperature for which the pilot thermostat is set, it passes air to supply the duct thermostat which in turn passes air to the valves on the heater coils. Thus the unit will deliver air at the temperature setting of the duct thermostat (usually 70 F). When the room temperature falls below the setting of the pilot thermostat the supply line is closed and the air is exhausted from the duct thermostat supply pipe. This prevents the duct thermostat from closing the heater coil valves, since it has no air, and maximum heating may be produced to raise the room to the desired temperature.

All indirect heaters are equipped with jet-type humidifiers and diaphragm valves which are controlled by hygrostats. These hygrostats contain an element which is sensitive to humidity changes and thus operate the necessary air valves.

The automatic louvre dampers for the fresh air intakes, recirculation ducts, and fan discharge ducts are equipped with damper motors which are controlled by pneumatic switches located on a switchboard. These switches pass or exhaust air to the damper motors which in turn are connected to open or close the damper louvres.



Section Showing Ventilating Ducts from Overhead Trunk System for Grand Salon

The operating element of the thermostat is a phosphor bronze vapor disc which is filled with a volatile fluid, the expansion and contraction of which, due to temperature changes, operates the valve mechanism of the thermostat.

Temperature and Humidity Regulation for the Pipe Organ

Accurate temperature control is necessary to keep the pipe organ in tune, because the expansion and contraction of the pipes through cold and heat will raise or lower the pitch. As the pipes are of different sizes and materials, this raising and lowering of pitch is irregular throughout the organ. The instrument will be out of tune unless the temperature is nearly the same as that at which it was tuned.

The temperature that is recommended normally is about 70 F. However, there are many cases where the temperature in an organ must be higher than this because of its position in the auditorium. The organ in the Scottish Rite Cathedral is above the ceiling of the auditorium and when the temperature at the floor of the auditorium is around 68 or 70 degrees, it will probably be 10 degrees more at the organ above. But if the organ is tuned at that temperature and if that is the normal temperature when the auditorium is right for the public to be comfortable, the organ will always come back in tune when this temperature is maintained. Organs are very often set higher than the floor of the auditorium and nearly always there is a higher temperature at the organ than in the auditorium below.

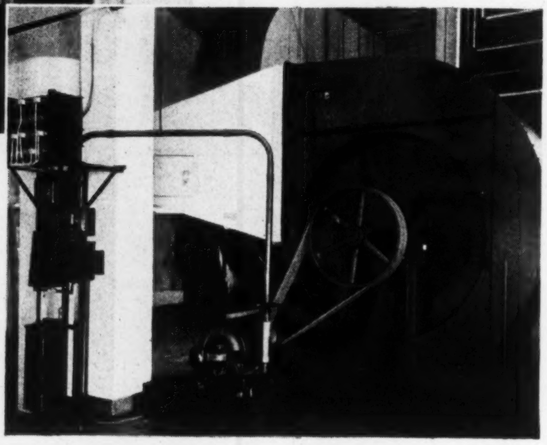
A certain amount of moisture in the air is necessary for an organ because it is built of wood and leather, glued together, and if it is allowed to get too dry, it will pull apart as will any other piece of cabinet work if it dries out too much. Therefore, moisture is supplied the organ chamber particularly where the auditorium is kept heated constantly to a normal degree throughout the entire winter season.

Humidification

Forty-five per cent relative humidity is maintained throughout the building. This is provided not only for the organ and for the comfort of the occupants but also for the preservation of the costly paneling and other woodwork.

The steam jet humidifier located in each section of the building consists of perforated steam pipes enclosed in vegetable fiber covering supplied with steam by a diaphragm valve.

Supply Fan for Grand Salon, Mounted on Sound Proof Platform and Operated With Silent Drive



Sound Isolation and Absorption

The prevention of the transmission of noises in buildings is an art that has been developed during the past twelve years. The use of equipment and materials for preventing noises in the Scottish Rite Cathedral was necessary because, due to its sturdy structure, it is a very resonant building.

Moving machinery was set on sound proofing platforms and many of the walls were covered with sound absorbing material. Sound transmission was also lessened by the canvas connections in the ducts.

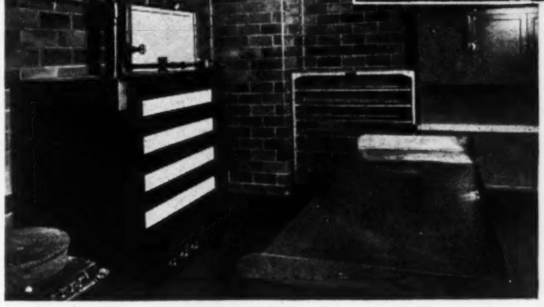
These were located every 30 feet and were made of 4 ply, 10 ounce canvas, shellacked.

An experiment was made by one of the operating engineers of the Cathedral by placing the motor-generator set for the projection booth on a platform which, in appearance, was like a sound proofing platform. The result was that the motor-generator set transmitted noise through the ducts to every part of the building. Part of the equipment that was installed for the machinery consists of a system of chairs and 2 by 4 cross members, together with a resilient cushion of felt material made of cattle hair, and part were set on 6 by 8 inch oak frames with two-inch cork mats between the frames.

Sound Absorbing Material Applied to Walls

The sound absorbing material applied to the walls also acts as a heat insulator. The prime object, however, in this instance was to decrease the general noise level in the room to prevent sounds from building up by successive reflections from hard surfaces. Ordinary building material such as wood, glass, plaster, cement, etc., absorb less than 5 per cent of the sound energy, while efficient sound absorbing materials may absorb 50 to 70 per cent of the energy. It is possible to calculate rather accurately the exact amount of material needed. The factors considered are the size of the room, absorption of the materials in the room and the use to which the room will be put.

One Section of the Main Kitchen. Kitchen Facilities Are Ample to Serve 4,000 People



Section of Bakery Showing the Gas Oven

In the banquet hall, for instance, 7,300 square feet of 1¼ in. sound absorbing material, with a coefficient of .70 were used; in the auditorium 2,900 square feet of $\frac{13}{16}$-inch sound absorbing material with a coefficient of .47 were used and in the grand salon, 3,700 square feet of $\frac{13}{16}$-inch material were used. The material utilized is a light, rigid tile, of cane fibers having 441 holes per square foot to increase its sound absorbing efficiency.

The usual defects of unsatisfactory acoustical conditions in auditoriums is that there is not sufficient absorption of sound in the walls, floors and ceilings to cause noise to die out in a short length of time. The lingering trail of sound from these reverberations prevents the hearers from being able to distinguish the syllables of speech and the notes of music. By the use of sound absorbing material, the length of the time which sound lasts can be shortened to the time most satisfactory to human ears.

Sound absorbing material has a value beyond that of rendering speech so that it can be understood and the music so that it will be pleasing. Reverberating sounds have a definite effect upon the nerves. The *Acoustical Society of America* went so far as to say that noise made inside of factories (or which penetrated them from the outside) lessened the efficiency of employes by 10 per cent. It is stated that heavy elevated traffic near an office may decrease the efficiency of employes, in some instances, 25 to 40 per cent.

Air Filtration

All entering air is filtered through a mat of copper expanded metal or copper mesh about two inches thick, immersed in oil. Filters are of the self-cleaning type and have capacities for handling about 190,000 c.f.m.

AIR FILTERS

	Capacity	Velocity	Size
Banquet Hall....	2—36,000 c.f.m.	460	13 ft. 3 in. x 11 ft. 6 in.
Grand Salon.....	1—38,000 c.f.m.	460	13 ft. 3 in. x 12 ft. 6 in.
Auditorium	1—42,000 c.f.m.	460	14 ft. 9 in. x 12 ft.
Main Floor......	1—34,000 c.f.m.	460	13 ft. 3 in. x 11 ft. 6 in.

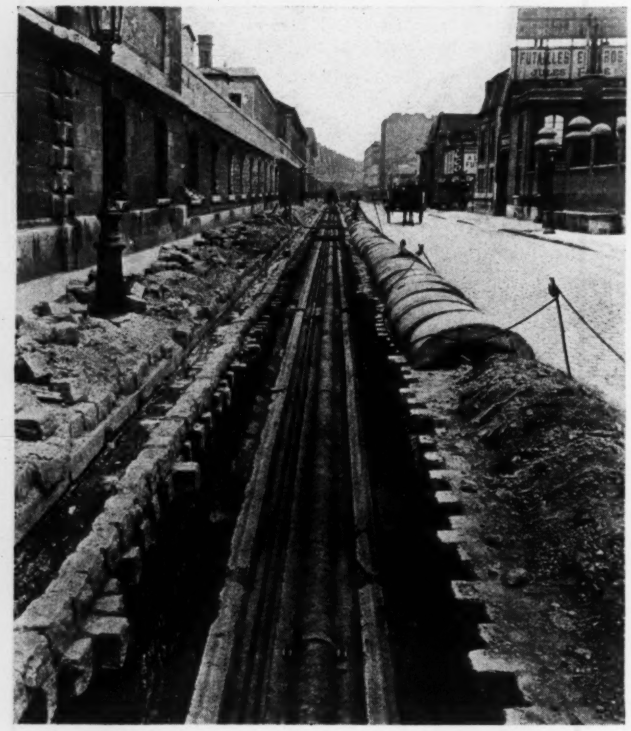
Water Supply

Water at 54 F pumped from a well 370 feet deep is used for the drinking fountains. In going through the pipes it picks up 4 degrees and is cooled again to 50 F by a one-ton refrigerating machine. A circulating ice water system is used for the water supply to the kitchen on a 2½ in. branch taken from the 4-in. main.

The water heaters supply the lavatories and showers, cross connected so that they can be used singly or together. Supply for showers and lavatories is carried at 120 F and for the kitchens at 180 F.

Steam Piping for New District Heating System in Paris[1]

By Phillippe Schereschewsky



VIEW OF LA RAPEE LINE IN RUE DE BERCY, GARE DE LYON DISTRICT, SHOWING PIPES IN PLACE BUT NOT INSULATED

DISTRICT heating, a utility service which has become very popular in the larger cities of America, will now be available in Paris according to the plans of the Compagnie Parisienne de Chauffage Urbain which was organized in 1928 and has the exclusive franchise in the city of Paris for the distribution of heat by steam or hot water. The enterprise is sponsored for the most part by the electric companies in the Paris district, the city government having taken no part in the raising of the capital of 80,000,000 francs ($3,140,000) which will be required. This article describes the general plan for the development of the service and the first steps which have been taken in its execution, but inasmuch as the actual operation of the system is only just starting no operating data is available.

The company has endeavored to find the districts most favorable from all viewpoints for district heating. It commenced by preparing statistics, house by house, of central heating installations[2]; also it has made a detailed study of the underground conditions, the subsoil in an old city like Paris being especially full of obstructions.

Load Different from American

The statistics thus collected showed that much less heat is used in Paris per square mile of ground area than in America. This is explained by the mild climate—the mean winter temperature in Paris being between 3 and 4 C (37.4 and 39.2 F)—and by the fact that the heating systems are designed for an inside temperature of 18 C (64.4 F), at an outside temperature of minus 5 C (23 F), and because the buildings are generally not as high as those in large American cities. Furthermore, even in those sections where central heating is most widely used, about one-half of the buildings are still heated by stoves. The amount of heat used throughout the winter in relation to the maximum load is also much less in Paris than is customary in American cities.

Boiler Plants Must be Outside the City

For aesthetic reasons the city absolutely forbids the construction of boiler plants not only in the heating districts but throughout the entire city. Consequently the steam supply must be obtained from points outside of the city, which involves the construction of large pipe lines from plants located at the outskirts, with consequent heavy expenditures which would not be necessary if plants might be located near the centers of load. The capital invested in underground pipes is therefore relatively much greater than in district heating systems in America where the heating plants are almost always located near to the points of greatest consumption.

Steam from Electric Plant

These considerations influenced the company to serve the west side of the city, taking steam from a suburban electric plant. The electric plant nearest the district of greatest heat consumption is that of Issy-les-Moulineaux, situated on the banks of the Seine about 500 meters (550 yards) south of the Paris city limits and three to four kilometers (1¾ to 2½ miles) from the district of greatest consumption. It is this electric plant which will feed the main network of heating mains. The maximum steam demand in the districts near the plant, which are mostly residential, will be about 1,102,000 pounds per hour. The plant contains high pressure boilers which must be supplied with distilled water and the district heating system is, therefore, designed so as to return the condensation from the consumers' buildings to the plant, where after filtering and degassing, it will be re-used in the boilers.

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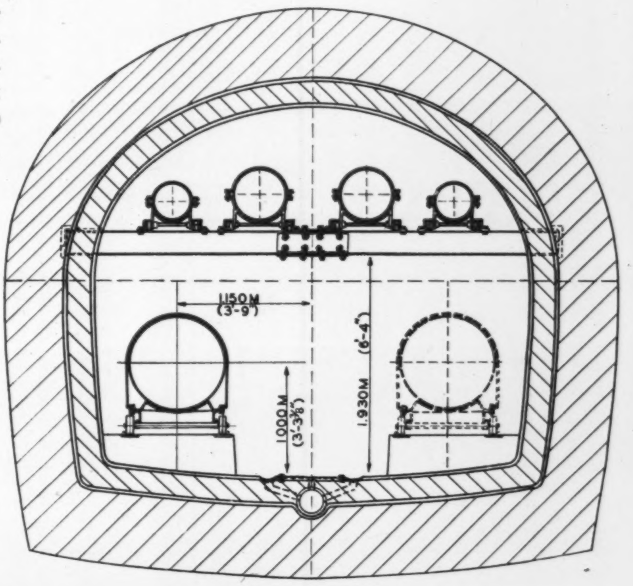
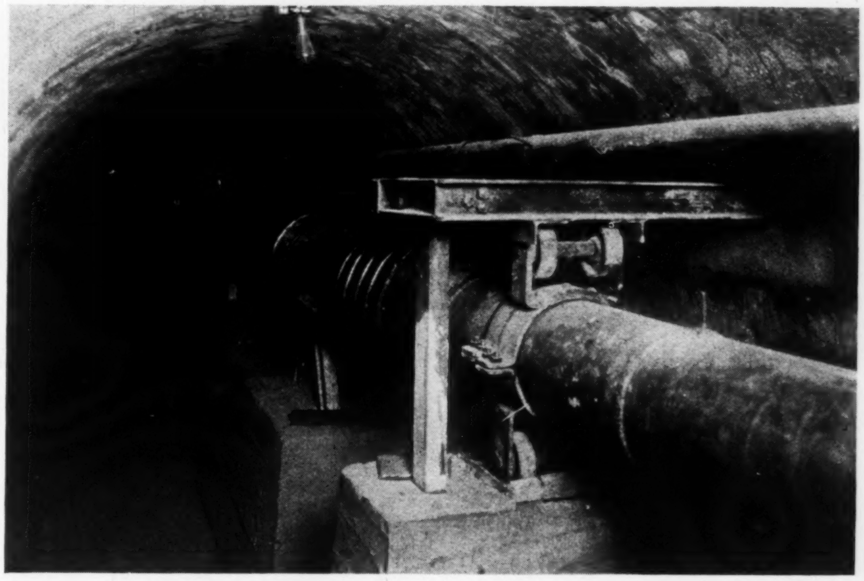
[1] Abstracted by J. H. Walker from an article prepared by M. Schereschewsky, Directeur de la Cie Parisienne, de Chauffage Urbain, for the Bulletin of the *National District Heating Association.*

[2] The term "central heating" as used here refers to heating by means of a single heater rather than by fireplaces or stoves. (Ed.)

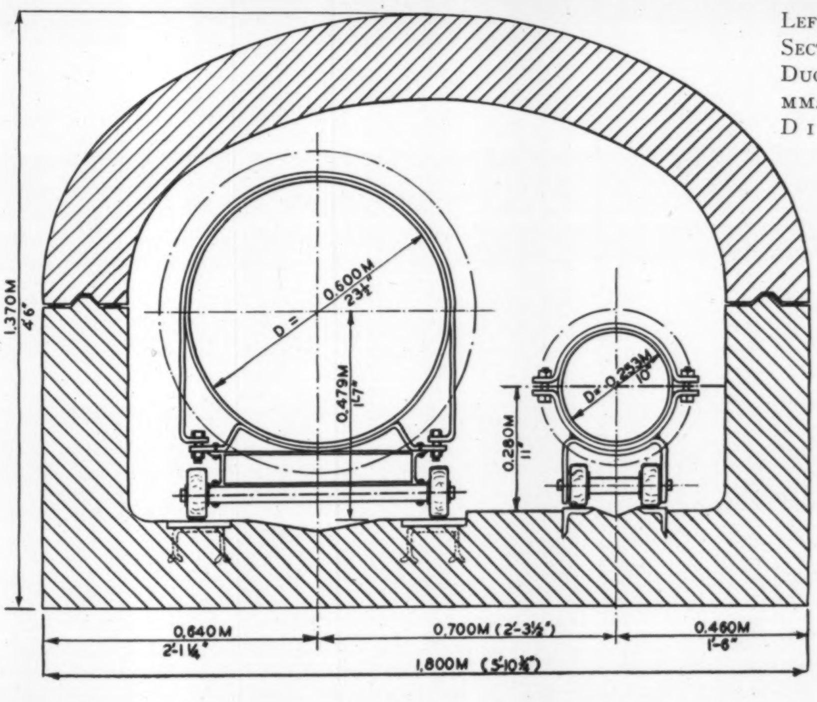
The total length of the main pipe line served by this plant will be 15 km. (9.3 miles) and its diameter will be about 800 mm. (31½ in.) at the start and will decrease in proportion to the distance from the central station. The total possible capacity of this line will be about 277,760 pounds of steam per hour and the hourly heating requirements of the buildings connected is on the average between 800 and 2,000 pounds of steam per hour.

Cross-Section of the Tunnel Between the Issy-Les-Moulineaux Power Plant and Auteuil Bridge





A View in the Old Metropolitan Cable Tunnel, Showing a Bellows Type Expansion Joint Applied to a 253 mm. (10-Inch) Diameter Main



Left — Cross-Section of a Duct for a 600 mm. (23½-In.) Diameter Main

The Conduits

The pipes will, in general, be installed in conduits which will not be open for inspection; however, for a distance of about 500 to 600 meters (550 to 600 yards) from the power house they will be installed in a tunnel in which men can walk.

This tunnel, located along the Seine in unstable ground and in some places below water level, will be of difficult construction, particularly because of the underground obstructions in this district. Generally speaking, the work of constructing the tunnels and conduits will be much more difficult and expensive in Paris than in the United States because the streets are not so wide and are already obstructed by various conduits, particularly sewers. The sewers of Paris are famous for their great size, which is due to their having been designed so as to be capable of inspection.

Experimental System an Aid

In addition to this principal system, the company has at present nearly completed an experimental system situated in the Gare de Lyon section 1 km. (⅝ mile) long. This line, constructed in co-operation with the firm of Rud. Otto Meyer, will be supplied with steam by the old central station of the Metropolitan Railroad and has just been put in operation. The performance of this experimental system during the winter of 1930-31 will be of great value to the company in settling a number of questionable points and establishing a rate for the sale of steam.

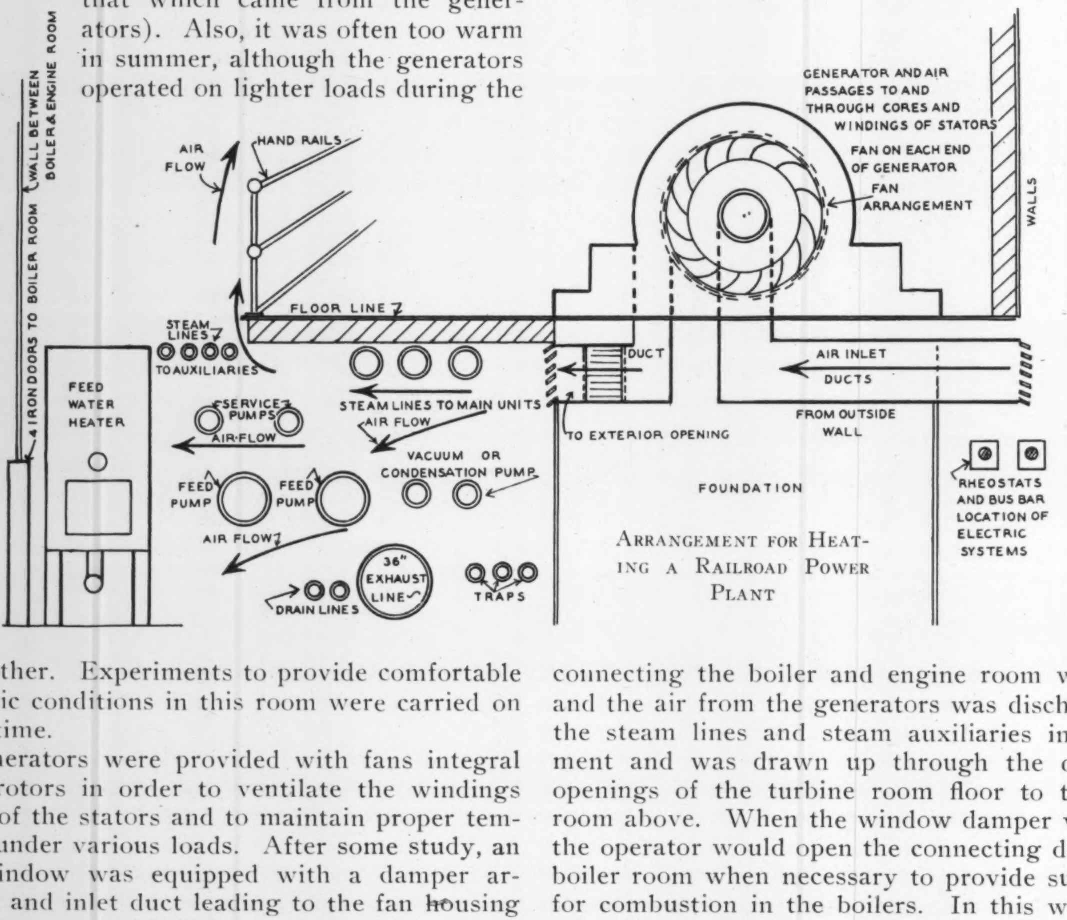
Detailed study of the district heating problem in Paris indicates that the financial returns from district heating can be satisfactory only if the steam is purchased at a very low price, principally because of the high capital charges resulting from the distance of central stations from the centers of load. This can be accomplished only by close co-operation between the district heating system and the producers of electricity in the neighborhood of Paris.

The steam intended for district heating can be relatively low in price if it is supplied from back pressure turbines (or better, extraction turbines because they can meet in a suitable manner the independent demands of the city for heating and electric energy, whatever the outdoor temperature may be.)

The system is expected to be ready for operation for the winter of 1931-32 and completed by 1932-33.

Air from Generators Heats Power Plant

The following method of heating and cooling a large railroad power station was used by the writer some years ago. Perhaps some other engineer may find this method of interest or may encounter conditions to which it may be applied. During the winter, the room enclosing the turbine generators was too cold for the operating crews as there were no radiators in the building and no heat was supplied (except that which came from the generators). Also, it was often too warm in summer, although the generators operated on lighter loads during the warm weather. Experiments to provide comfortable atmospheric conditions in this room were carried on for some time.


4 IRON DOORS TO BOILER ROOM
WALL BETWEEN BOILER & ENGINE ROOM
AIR FLOW
HAND RAILS
FLOOR LINE
STEAM LINES
TO AUXILIARIES
FEED WATER HEATER
SERVICE PUMPS
AIR-FLOW
STEAM LINES TO MAIN UNITS
AIR FLOW
VACUUM OR CONDENSATION PUMP
FEED PUMP
FEED PUMP
AIR FLOW
36" EXHAUST LINE
DRAIN LINES
TRAPS
GENERATOR AND AIR PASSAGES TO AND THROUGH CORES AND WINDINGS OF STATORS
FAN ON EACH END OF GENERATOR
FAN ARRANGEMENT
WALLS
DUCT
AIR INLET DUCTS
FROM OUTSIDE WALL
TO EXTERIOR OPENING
FOUNDATION
RHEOSTATS AND BUS BAR LOCATION OF ELECTRIC SYSTEMS

Arrangement for Heating a Railroad Power Plant

The generators were provided with fans integral with the rotors in order to ventilate the windings and cores of the stators and to maintain proper temperatures under various loads. After some study, an outside window was equipped with a damper arrangement and inlet duct leading to the fan housing of the generators. A discharge duct led from the lower section of the housing to the basement of the turbine room, the air passing over steam coils. It was so arranged that the air could be discharged to the interior of the building or to the exterior as desired. By manipulation of the window damper and by increasing or decreasing the opening, comfortable conditions were maintained during both summer and winter.

The window was controlled manually and experiments were made with openings of various sizes. By decreasing the opening, the generators were caused to rise to a desirable temperature in cold weather. When the outside temperature was sufficiently cold to warrant it, the window damper as well as the door connecting the boiler and engine room were closed and the air from the generators was discharged over the steam lines and steam auxiliaries in the basement and was drawn up through the observation openings of the turbine room floor to the turbine room above. When the window damper was closed, the operator would open the connecting doors of the boiler room when necessary to provide sufficient air for combustion in the boilers. In this way, the air inlet for combustion was sometimes closed to conserve heat.

This method worked satisfactorily and comfortable conditions were maintained at all times.—*Thomas J. Brett.*

Piping to Heat

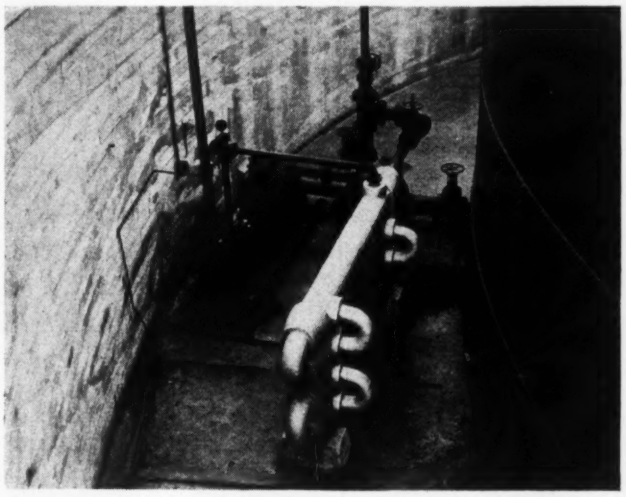
High Viscosity Oil for Gas Manufacture

with Exhaust Steam

RECENTLY the heating of heavy oils in storage tanks has been successfully accomplished in those having no coils by the use of a heating device set up adjacent to the tank. By this means any available tank can be utilized. The idea of a thermal siphon (such as a gas fired water heater depends upon to heat the water in an adjacent storage tank, or the operation of an automobile radiator) was applied to these large tanks with the exception that instead of depending upon a circulation due to rising temperature, the flow was maintained and regulated by a pump in the system. The heater employed was one of the type now offered for use in the industry, consisting of a series of chambers ranging above each other and filled with a large number of brass tubes of small diameter through which the oil flows in an upward direction while the steam for heating surrounds the tubes, flowing counterflow to the oil to the outlet valve or trap at the bottom. The pump was connected between the bottom of the tank and the heater, and the discharge from this heater was arranged so that when no oil was required in the plant the heated oil would return to the tank, but when oil was needed it was immediately available and during these periods of using the return to the tank was only the difference between the rate of pumping and the amount being used.

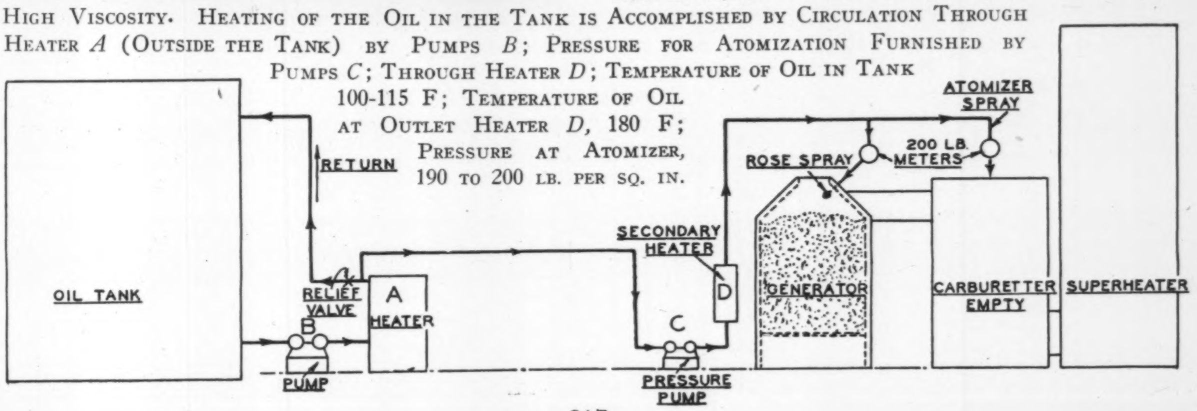
Where the use is more intermittent as, for instance, in the occasional filling of a car or other container, the total capacity of the pump can be returned to the tank. Another feature in favor of this type of heater is that due to its small size and great efficiency a close control can be kept on the temperature of the oil in the tank, so that excessive heating will not occur with the corresponding loss of heat by radiation. In the event of trouble in this type of heater it can be shut down and drained in a short time and immediately repaired, or, if a similar unit is maintained in reserve, it can be replaced in a few hours and normal service restored.

When it was first decided to try this type of heating device some old unused heaters (which were standing in reserve, although known to be less efficient than those recently developed and now on the market) were used to prove the possibility of such a system. A tank containing several hundred thousand gallons of heavy oil was connected to these heaters and a pump, and exhaust steam at six pounds pressure was used for heating. Results were so successful that the experimental hook-up was used throughout last winter to keep the oil in this tank fluid. Following this, the other tanks requiring such heating were equipped with the more efficient type mentioned previously. It was found in the experimental apparatus first installed that a heat transfer of 13 Btu per hour per sq. ft. occurred when circulating about 8,000 gallons per hour, and in practice it was possible to maintain a temperature in the tank between 120 and 140 deg. F, these changes being due to the lower temperature of the barge loads of oil which from time to time were pumped into the tank. All of the large tanks, ranging from 750,000 to 1,000,000 gallons capacity, which contain heavy oil, are now equipped with this external heating system, which is working so efficiently that the system has been adopted as part of our standard for the future construction of tanks destined for this service.—*Harry E. Hodgson,* Philadelphia Gas Works Co.



Left—Heater Installation on a Million-Gallon Storage Tank

Below—Arrangement of Tanks, Heaters and Piping to Water Gas Sets, in Use in a Plant Capable of Making 45,000 Cu. Ft. of Gas Per Day, Using Oil of High Viscosity. Heating of the Oil in the Tank is Accomplished by Circulation Through Heater *A* (Outside the Tank) by Pumps *B*; Pressure for Atomization Furnished by Pumps *C*; Through Heater *D*; Temperature of Oil in Tank 100-115 F; Temperature of Oil at Outlet Heater *D*, 180 F; Pressure at Atomizer, 190 to 200 lb. per sq. in.



The Lakeside Press Air Conditioned for Comfort of Personnel

By J. H. Bailey*

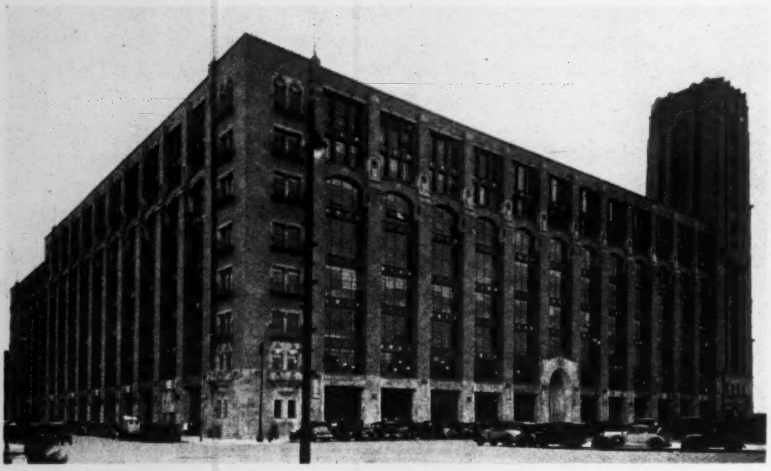


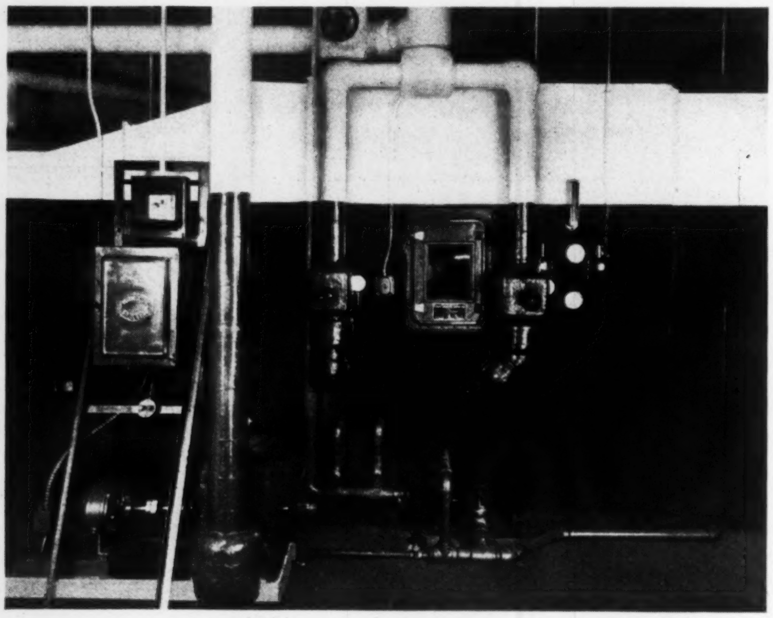
FIG. 1—THE LAKESIDE PRESS (PLANT AND OFFICES) AT CALUMET AND TWENTY-SECOND STREETS, CHICAGO. THE MANAGEMENT OF THIS LARGE CONCERN RECOGNIZES THE VALUE OF CONDITIONED AIR IN INCREASING THE EFFICIENCY AND COMFORT OF OFFICE EMPLOYES

COOLING and air conditioning for the purpose of providing comfortable atmospheric surroundings for occupants of modern offices is increasing rapidly in popularity. Business men recognize today that proper air conditioning for employes affects operating costs favorably and is as beneficial as is controlled weather for the processing of materials, which has been done for many years. Air conditioning for comfort has made rapid strides in theaters, department stores, restaurants and similar institutions. To insure maximum cleanliness and minimum investment in the air conditioning plant, tight building construction (double windows, tight ceilings and floors) is helpful. The use of nonabsorbent, dustless materials for floors and shades is also of assistance.

The air conditioning system recently installed in the plant of the R. R. Donnelley and Sons Company (The Lakeside Press) at Calumet and Twenty-second Streets, Chicago, was designed to provide comfortable and clean working conditions for its executives and office employes. Two sets of air conditioning apparatus are used to maintain an inside temperature of 80 F and 50 per cent relative humidity in the summer and a temperature of 72 F and 35 to 40 per cent relative humidity in winter, as part of the offices are located on the first floor and part on the eighth. A view of the apparatus (located in the basement) is shown in Fig. 2.

One complete dehumidifying equipment serves the offices of the R. H. Donnelley Corporation and another set serves the R. R. Donnelley and Sons Company, also supplying their library, exhibition hall, etc. Each set of air conditioning equipment consists of preheaters, oil-type air filters, dehumidifiers, re-heaters, fans, etc. The refrigeration machine (shown in Fig. 3), which is of the centrifugal type, supplies refrigerated water to the two dehumidifiers. The approximate capacity of this

FIG. 2—A VIEW OF THE AIR CONDITIONING APPARATUS, LOCATED IN THE BASEMENT OF THE EIGHT-STORY BUILDING



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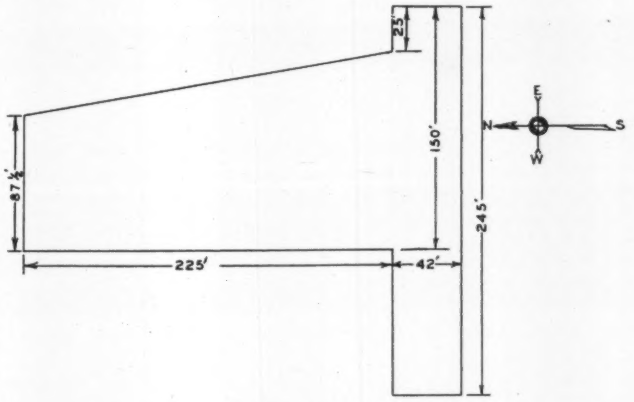
*Engineer, Carrier Engineering Corp., Chicago, Ill.

machine is 200 tons. The approximate capacity of the combined air conditioning systems is 110,000 cubic feet of air per minute.

The R. R. Donnelley office region is located on the top floor of the building. The purchasing, clerical and sales forces are located in the older portion of this region and designated as Building 8-C. This particular space was originally used for paper storage, but plant rearrangement made it highly desirable to locate part of the office force on this level, since the executive region was to be in Building 8-D, which was to be constructed.

In the design of the air conditioning system many problems had to be solved, and with the able assistance of Charles G. Atkins, the consulting engineer, and E. G. Greenman, the engineer for the owners, a scheme was evolved which met all of the requirements. The unsymmetrical floor plan of the building made careful study essential in order that a desirable temperature distribution could be obtained. An idea of the shape of the air conditioned areas described in this article and their approximate dimensions is given by the sketch above.



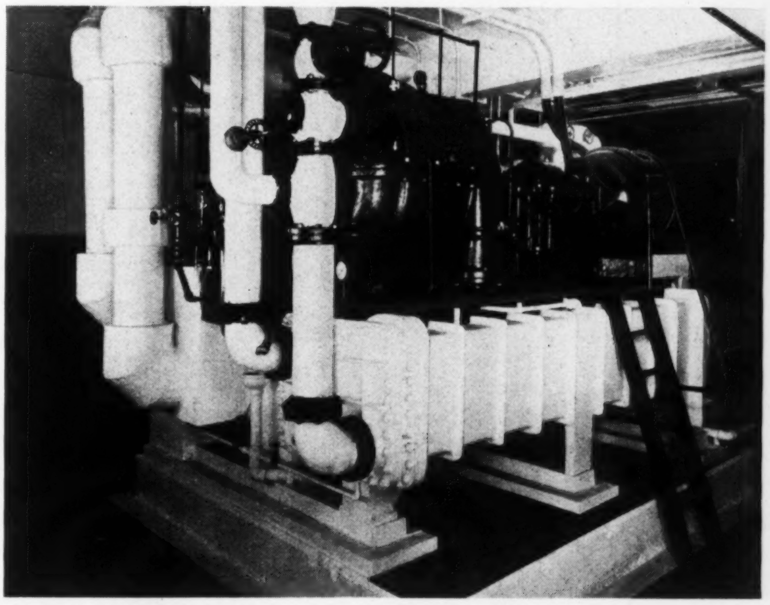


Fig. 3—The Refrigeration Machine Supplies Refrigerated Water to Two Dehumidifiers

Then, too, the installation of an air conditioning system in an existing building always calls for considerable ingenuity. For instance, on this job an abandoned stack 120 ft. high was partitioned to form the supply and return risers from the basement to the eighth floor. This shaft connects (at the floor line) with the suction system beneath the floor and at the ceiling line with a net-work of supply ducts serving the area through grilles placed in artificial beams which were built on 25 ft. centers the length of the building.

Fig. 4 is a construction photograph and shows clearly the air distribution system overhead and the exhaust or return air system between the main floor and the false floor twelve inches above. Fig. 5 shows a view of the same office after completion, taken from a slightly different location, but showing the same ducts.

The ceiling above the beams is covered by a sound insulator, completely concealing the ducts. Air velocities in the underfloor system are approximately 375 f.p.m., in the supply and return shafts about 1800 f.p.m., and the branch ducts were designed for 600 f.p.m.

The hollow tiles placed on end (see Fig. 4) serve a double purpose; first, as a floor support, and secondly, they were so arranged in line as to serve as separate return ducts, for the maximum distance some of the return air has to travel is approximately 375 ft. to the fan inlet, whereas other return registers are much nearer the apparatus. Therefore, since uniform distribution is essential to maintain uniform temperature, it was a necessity to have each and every supply and return grille do its part of the work no matter how far removed from the apparatus. Each of the tile alleys could be restricted to create positive pull. The sizes of these "tile tunnels" (tile on the sides and concrete on the top and bottom) were calculated from special resistance data applying only to such peculiar construction.

Areas most difficult to ventilate are those near windows, because of the cold in winter and heat in summer. In order to insure positive circulation of heated or cooled air along the windows, 50% of the return grilles were placed in the floor immediately adjacent to the glass.

Separate lateral duct systems supply conditioned air to the east side and west side of Building 8-C. This is of paramount consideration, as the structure is practically on the shore of Lake Michigan, the east side being exposed; this zoning made it possible to obtain accurate control of room conditions. It often occurs that on one side of this building heat will be required, while on the other side the air will be delivered at or below room temperature. In the latter case, of course, it will be cooling. Room thermostats are located throughout this area and control the admission of steam to corrective heaters and operate the partial area dampers for each respective area. Double window construction and careful design of supply and return registers made it possible to obtain accurate temperature control.

Fig. 6 is a view of the exhibition hall and is indicative of the beautiful type of construction that predominates throughout the plant and offices. It is typical of the two two-story rooms supplied by the air conditioning system. To the east of this hall are numerous executive offices, the same general control design being used in that area as in the main office space. However, in the executive region, it was possible to incorporate the air conditioning system in the original building plans. In each case the air is delivered at the ceiling and the return air is removed at or near the floor line. Special thought was given to the areas and locations of all grilles in order to eliminate any possibility of drafts; low supply velocities, under 150 f.p.m., are instrumental in securing such results.

R. R. Donnelley & Sons Company specialize in high-grade printing work. In addition to providing ideal atmospheric conditions for the executives and employes, the air conditioning system will assist in the preservation of samples of printing for the benefit of customers and the sales force.

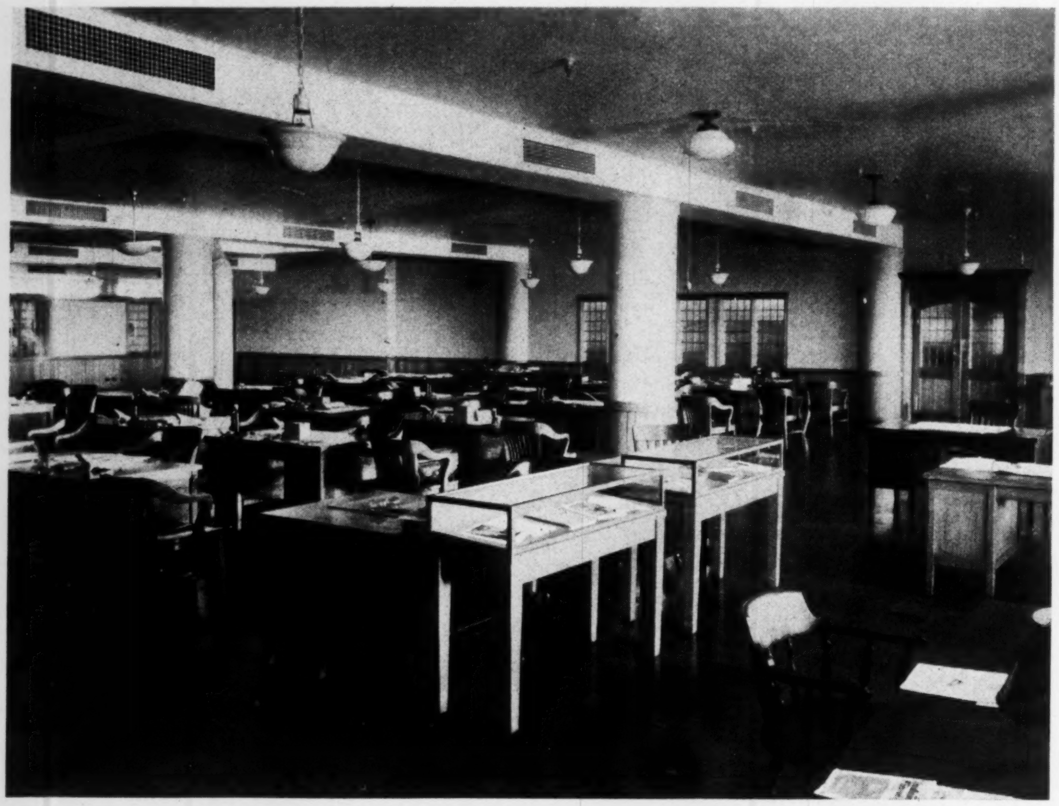


Fig. 5— (Above) A View of the Same Room Shown in Fig. 4. Note the Air Outlets in the Beams

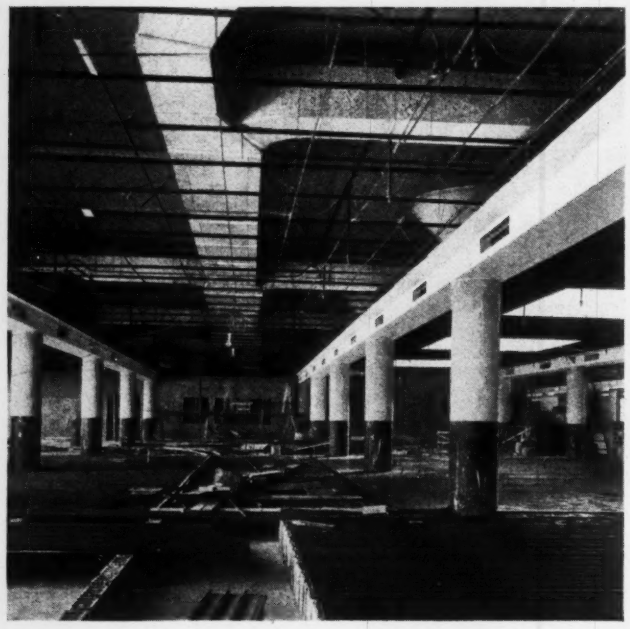


Fig. 4— (Right) A Construction Photograph Showing the Air Distribution System Overhead and the Return Air Ducts Between the Main Floor and the False Floor. The Hollow Tiles Serve as a Support for the False Floor and Also as Separate Return Ducts, as They Are Placed in Line

It is worthy of note that the final decision to install air conditioning for the offices of the Lakeside Press involved an installation in an existing building. Obviously the cost of an installation is higher under such circumstances—but the decision to proceed indicates a thorough conviction that even at a material increase in cost the step would be a sound one. It further indicates that a great many buildings—and particularly offices occupied by the prospective owner of an air conditioning system—can be provided with this modern service on an economically sound basis, without waiting for a new building or new quarters where air conditioning is already provided.

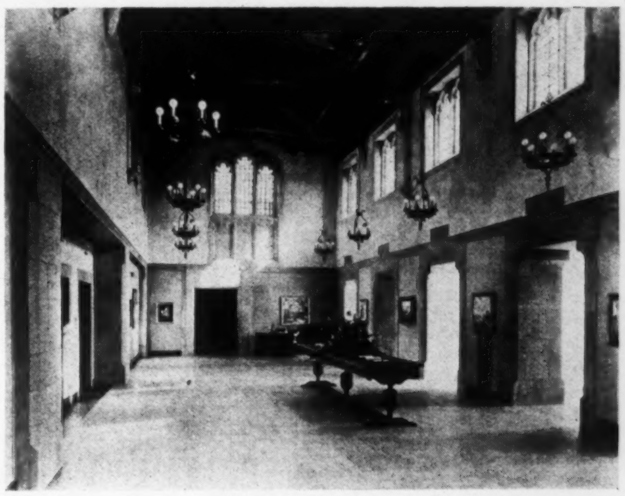
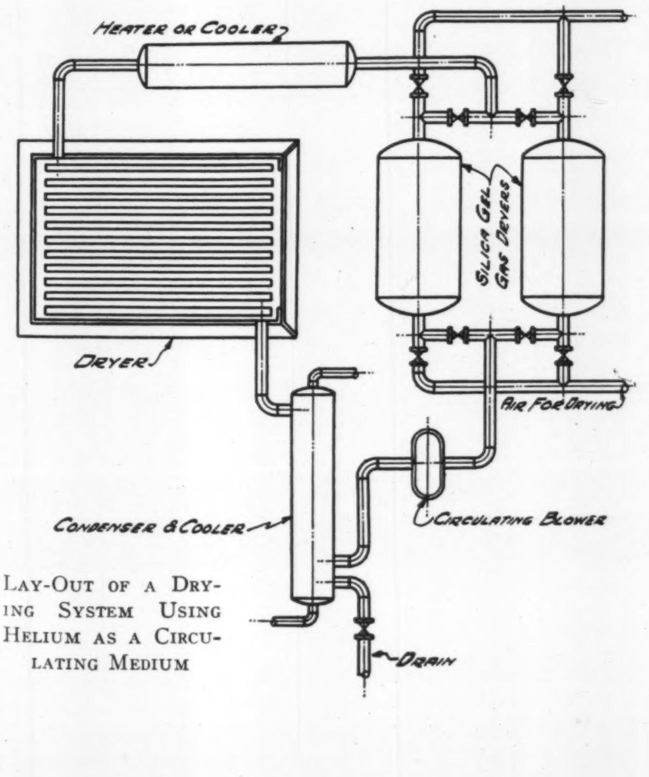


Fig. 6—The Exhibition Hall on the Eighth Floor, Also Air Conditioned. The Construction is Typical of the Quality Throughout the Entire Structure

Helium as a Circulating Medium in Drying Operations

THE property of chemical inertness possessed by helium, coupled with its high heat conductivity and other peculiar properties, makes helium of especial interest as a circulating medium for the drying of a great many substances. The use of helium for this purpose will give quick action, will be economical in operation; the equipment used has simplicity and is safe to operate. Production of an excellent product and the recovery of solvents are two other features of drying by this method.

Among the many products that can be dried to advantage by the use of helium are fruits, vegetables, meats and fish, cereals, milk powders, casein, yeast, gelatin, albumen, starch, sugar, glue, organic extracts, rubber, electrical insulating materials, soaps, explosives, and electrical equipment such as armatures, instruments, etc.



Lay-Out of a Drying System Using Helium as a Circulating Medium

Other delicate organic and inorganic materials that would be adversely affected either by high temperature or by contact with any active gas such as air, steam, carbon dioxide, etc., can be dried to advantage through the use of helium.

It is well-known that liquids in contact with fixed gases have a higher vapor pressure than when in contact only with their own vapor*; consequently materials can be dried quickly in contact with fixed gases where circulation of the fixed gases is sufficiently rapid to remove the vapors. The circulation of a fixed gas through a chamber drier also sweeps out the vapors rapidly.

Apparatus Used

The illustration shows the layout of an installation in schematic form for accomplishing the drying of products in an atmosphere of helium. The helium used in this drying is kept in a closed cycle and the helium circulated by means of a blower. The helium, before passing to the heater or cooler, as the case may be, and thence to the drier, is completely dehydrated in one of the silica gel gas dryers. The helium then passes to the heater or cooler where, depending upon the characteristics of the product to be dried, the temperature is raised or lowered to the most desirable point and then passed over and around the product contained in the trays in the drier.

From the drying chamber the helium, with the moisture absorbed from the material, is drawn through the condenser where the excess moisture is condensed and drawn off and the helium, saturated with the water or solvent, then passes again through the circulating blower and dryers.

Where the product being dried contains solvents other than water, these solvents will be absorbed in the gel from which they may be recovered by appropriate means. In the recovery of solvents from products being dried, the silica gel may be replaced by activated charcoal or other absorbent materials. Two of these gas dryers are placed in parallel, one of them being reactivated while the other is being used for the drying of the gas.

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*Euken, Jett and La Mer—Principles of Physical Chemistry, Page 149

Moisture Flow Data Needed to

Choose Low Humidity Equipment

THE POSSIBILITIES of low humidity for drying electrical equipment, designed for low voltages, and for treating lung, throat and nose diseases in hospitals has been pointed out frequently in recent engineering literature. That keen interest has been aroused in medical circles is evident from the fact that a number of hospitals already have installed air conditioning plants, and that small units for hospitals have also appeared. The latter indicates that there is a demand for such service.

Low humidities involve certain problems not encountered in the "high humidity" field. One of these problems is preventing the infiltration of moisture to the conditioned room from the surrounding atmosphere.

Practically no data can be found on the infiltration of moisture through various building materials. This makes it difficult to determine the additional amount of equipment which must be provided, so far as capacity is concerned, to handle this moisture. When the relative humidity is practically the same inside and outside of the conditioned room and the air pressures are uniform, it is evident that no "flow" of any appreciable amount of moisture will occur. Where fairly large variations or differences occur in these two factors it is evident that nature will make a strong effort to balance the relative humidity by a movement of moisture towards the space where low humidities prevail. It naturally follows that if the materials used in the construction of the room are hygroscopic, and if no attempt is made to seal the small pores, the only barrier to a flow of moisture through the walls, ceiling and floor will be the resistance set up by cells in the material. In other words the smaller the cells the slower the moisture will travel through the material. Take a brick of any type, obtain its weight and then, after soaking it in water, weigh it again. The large amount, proportionately, which the brick can pick up is a good example of how much moisture can travel, or infiltrate, through building materials of like porosity.

How Moisture Enters Rooms

When it is considered that moisture will also enter the low humidity room in the materials which enter, on the skin and in the clothes of the persons who pass into the room through the doors and also by means of the infiltration of air around doors and windows, it is evident that the engineer has a real problem in calculating the requirements for air conditioning when there is no experimental data on this important source of moisture, infiltration.

An Example

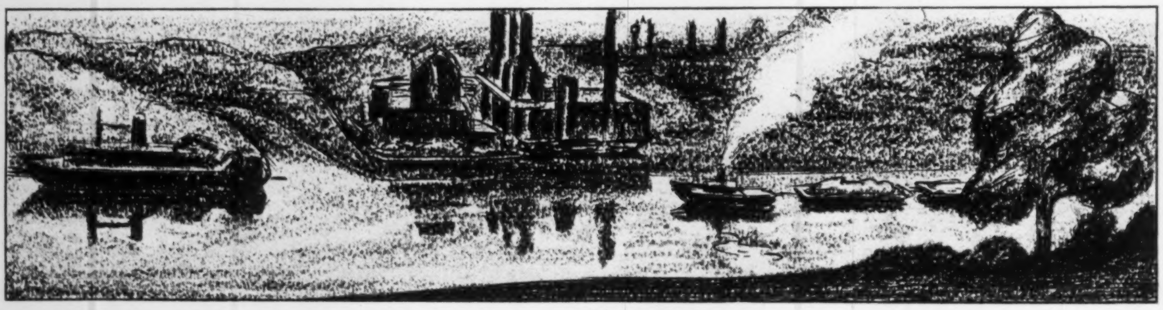
What does this flow of moisture mean? To enable the reader to visualize the answer to this question a practical example will be given.

A room of 90,000 cubic feet, with brick walls, a concrete floor and a rough plastered ceiling, had a temperature of 110 F, with an air motion not faster than 100 f.p.m. The surrounding space had no mechanical air motion and its air temperature was about 70 F. The relative humidity desired in the first space was ½ per cent, while the surrounding space was never lower than 20 per cent. After the walls and ceiling alone had been thoroughly water-proofed, a ½ per cent relative humidity was easily maintained but, previously, the relative humidity could not be dropped below 2½ to 3 per cent. Ninety per cent of the air was recirculated. The remainder, or 800 cubic feet per minute, was dried. If the waterproofing had not been used the amount of air dried would have had to be increased to over 4,000 cubic feet per minute or the capacity of the equipment would have had to be raised. This infiltration of moisture amounted, actually, to about 11 lb. of water an hour. Furthermore, it is certain that moisture was also flowing through the floor, which was not waterproofed, and that the capacity of the equipment might have been lowered to some extent if this last item of moisture had been eliminated.

Need for Research

It is evident that considerable need exists for research data on this subject of moisture "flow" or infiltration through building materials under various conditions of relative humidity and of air pressure. Such information would enable the "low humidity" engineer to figure the economic balance as to just how much waterproofing, sealing of cracks and tightening of door closures is needed on any particular job. Furthermore, it is evident that this information would be of value in the high humidity field also, where the lower humidities of course prevail in the atmospheres surrounding conditioned rooms.

In the meantime those who use low humidities apparently should waterproof all walls, ceilings and floors thoroughly and should use other precautions to prevent the infiltration of air. A valuable device for keeping down air losses, where considerable movement through doors occurs, is a corridor provided with two sets of air-locked doors.—*Malcolm Tomlinson.*



Trunk Ducts for the Fan Blast System of Warm Air Heating

By Platte Overton*

THERE are two systems of trunk ducts that the designer of direct-fired hot blast systems will encounter. One is the round or square galvanized iron trunk duct with branches proportioned to resistance, and the other the concrete or extended plenum system. With the concrete trunk or extended plenum system, the duct is generally carried through the same size. It may vary slightly toward the end and be made smaller, but where installed under corridors it is uniform in width, and the bottom is pitched toward the heaters. If the duct is large and well proportioned, a relatively equal pressure per sq. ft. of surface is maintained. In ducts up to 200 ft. long, there is very little difference in the pressure per sq. ft. at the near or far end of the plenum.

Out of this chamber near the top, openings lead to the warm air risers. It would be practically impossible to design such a duct so exactly that no volume dampers would be necessary; therefore, their installation is recommended.

The extended plenum may be designed for a velocity of 1200 to 1800 f.p.m. The openings from the plenum may be designed for a velocity of 500 to 600 f.p.m., but no hard or fast rule can be applied here as the velocity varies in the near and far end of the plenum, and the openings on the far end should be favored a little.

Two or more fans may discharge into the same duct, and with two or more ducts down through the building, the ends should be joined so that the excess from one duct may equalize with the other.

Individual Temperature Regulation

Where individual temperature regulation for the various rooms is required, a double duct is generally required as shown in Fig. 1. The tempered air at 60 to 70 degrees, is carried in the lower duct and mixed with air from the warm air plenum, through double mixing dampers thermostatically controlled. It will be found that the tempered and warm air plenum will equalize their own pressures, as the closing of the dampers in the warm air plenum would release the pressure in the tempered air chamber.

The writer has experienced cases where the plenum being extra deep, 8 to 12 ft., and not too long, say 50 to 75 ft., no partition was necessary between the tempered air and the warm air plenum due to the stratifying of the air at different temperatures (Fig. 2). In such cases a galvanized iron duct is run down inside the chamber with a warm air opening at the top where the mixing damper is located. The tempered air is taken into the duct at the floor line. This type of system is not generally recommended, but is giving fair results in several schools.

Heat Loss in Concrete Ducts

One hears the claim that there is an excessive heat loss in concrete ducts. This is not literally true. When the supply fan is shut off, say at night, the heat throughout the concrete walls is given up to the air in the chamber, and maintains a gravity circulation that carries this heat up to the rooms. This has been established by actual test. This test was taken on a large Wisconsin school with approximately 100 rooms. The distance from the heaters to the most remote room is 325 ft. With the fan shut down at 9:00 P. M., the temperature in this room dropped only 18 degrees until 6:30 A. M., when the fan was again turned on, and this with a 15 mile an hour wind blowing on the room and an outside temperature of

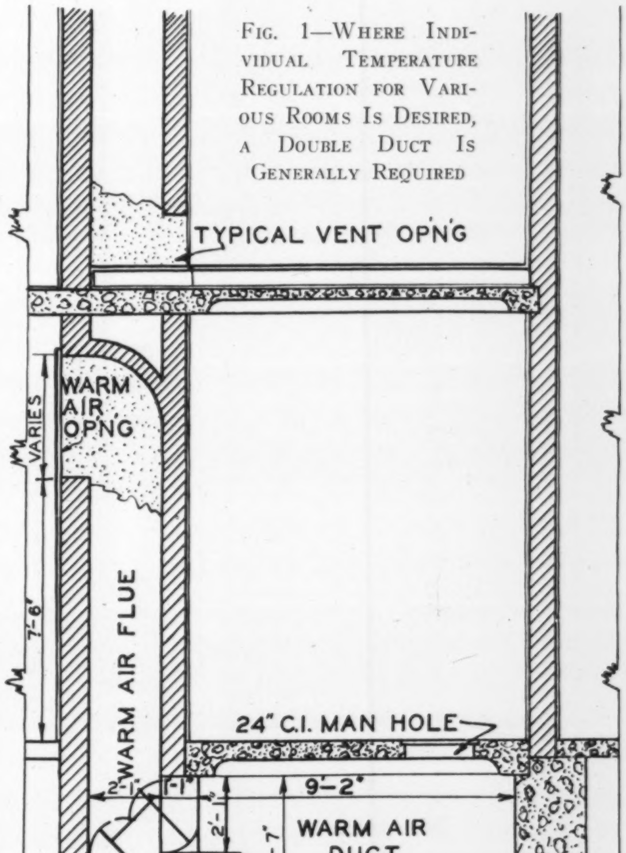


FIG. 1—WHERE INDIVIDUAL TEMPERATURE REGULATION FOR VARIOUS ROOMS IS DESIRED, A DOUBLE DUCT IS GENERALLY REQUIRED

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* Consulting engineer, Chicago, Ill.

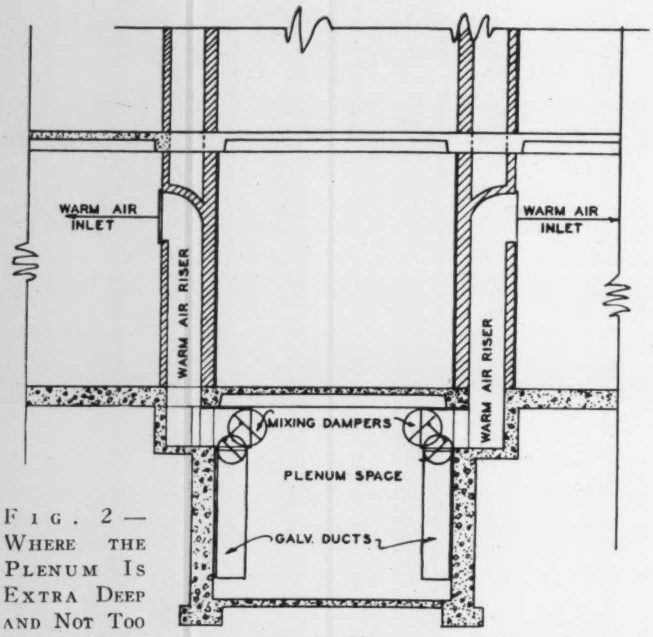


FIG. 2—WHERE THE PLENUM IS EXTRA DEEP AND NOT TOO LONG, A PARTITION MAY NOT BE NECESSARY, DUE TO STRATIFICATION OF THE AIR

7 deg. below zero. A thermometer in the room registered 70 deg. by 7:00 A. M.

Friction Loss

Regarding the friction loss in these ducts, they were found so low that the testing engineer would not believe the gage, and rechecked them several times in an attempt to prove the gage wrong. With 70,000 c.f.m. in a duct 250 ft. long, and a velocity of nearly 2,000 ft. per min. for half that length, the static reading was less than $\frac{1}{4}$ in.

The concrete extended plenum system is not recommended where intermittently used as in a church or public hall, or even in schools if located south of a line drawn between latitude 35 deg. and 40 deg. In churches used only a few hours a week, the services would be over before the concrete ducts were well warmed up.

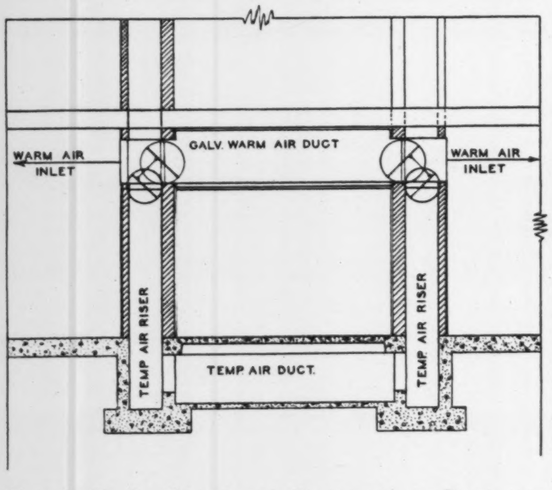


FIG. 3—IT IS SOMETIMES PRACTICAL TO RUN THE WARM AIR TRUNK DUCT ON THE CEILING OF THE FIRST FLOOR CORRIDOR AND THE TEMPERED AIR DUCT ONLY UNDER THE FLOOR

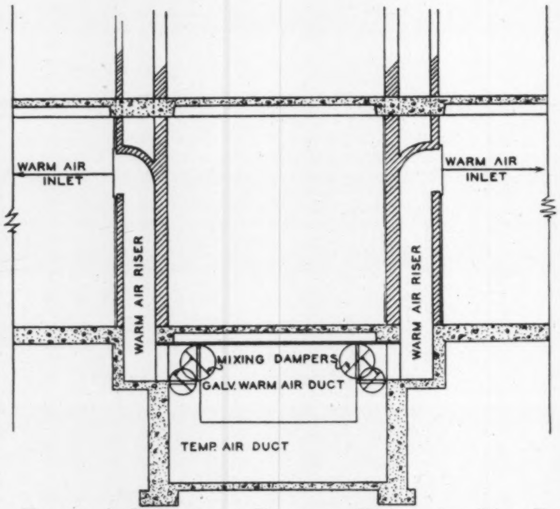


FIG. 4—A GALVANIZED DUCT FOR WARM AIR MAY BE RUN DOWN THE CENTER OF THE CONCRETE DUCT FOR TEMPERED AIR

Design of Concrete Ducts

Concrete ducts or extended plenum systems should be designed large and deep, and as straight as possible. Sharp turns are to be avoided, and where they are necessary, deflectors must be installed. They should be provided with lights and drains, and washed out with a hose when necessary. Steam mains, or water supply pipes, may be carried in them but they should contain no soil pipes or sewers.

It is sometimes practical to run the warm air trunk duct on the ceiling of the first floor corridor, and the tempered air duct only under the floor, as shown in Fig. 3, in case the ceiling is not high enough to permit two galvanized ducts, one for warm air and the lower duct for tempered air. A galvanized duct for warm air may be run down through the center of a concrete duct used for tempered air (Fig. 4), but if the distance is too great, the tempered air will have a tendency to over heat.

Design Velocities

The individual duct system is almost invariably designed to velocities. State codes in some cases fix these velocities by rule. Such a system is shown in Fig. 5. They vary from 650 to 800 f.p.m. Volume dampers are provided for each run of duct, and those in the shorter runs are set to equalize the resistance with the longer ones. It is possible to design each run to the resistance (arbitrarily taken), but this method must be used with caution for, while theoretically the only correct one, practically it may cause trouble if unforeseen conditions make it necessary for the erectors on the job to change the course of a duct or add more elbows.

The designer of the individual duct system should use judgment when designing ducts to velocities, and the extremely long run, or that with many elbows, should be favored with extra size. This holds true with the very short ones with no elbows, and they may be slightly reduced in area.

In the so-called combination system, where the fan runs intermittently, and the warm air is carried to the rooms by gravity in mild weather, the individual ducts should be designed for a velocity of 500 f.p.m.

Galvanized individual warm air ducts sometimes offer a problem when the question of cleaning is confronted. They should always be equipped with doors, or slides, but if they are long, and closely nested, correct cleaning is frequently difficult and if they are installed above a suspended ceiling it is more so.

The question of temperature drop in the individual ducts is not important unless they pass through the attic space, or some area where the heat loss would be excessive, or would be lost to some space where it could not be utilized to heat the space through which the ducts pass.

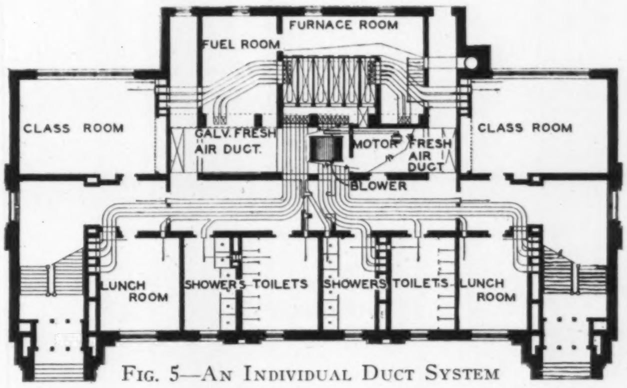


Fig. 5—An Individual Duct System

The throat of all elbows up to 12 in. in diameter should have a radius of 12 in., and those over 12 in., a radius equal to the width of the duct. Design the plenum chamber as large as practically possible. This makes it possible to maintain an equal pressure throughout the chamber and an opening 12 × 20 inches will get its quota of air even if located next to an opening 20 × 40 inches.

The galvanized trunk duct for use in direct-fired hot blast systems is designed for equal resistance per lineal foot. In designing such a duct it is a good plan to establish an overall resistance for the duct and work to it. In a well designed duct of this type, no volume dampers should be required, but very few engineers have the confidence to leave them out. Practical tables for proportioning trunk ducts and branches for equal resistance per lineal foot are obtainable, but they are based on round ducts. These may be changed to rectangular ducts by use of the table giving the circular equivalents of rectangular ducts for equal friction per lineal foot.

Piping systems for fan blast, direct fired installations are seldom complicated.

It is not good practice to blow downward into a warm air tunnel, and this should be done only where unavoidable. The direct-fired heaters should, wherever possible, be lower than the extended plenum or galvanized trunk ducts. A gravity service over the heaters when the supply fan is idle is important. All direct-fired fan blast heaters must be placed on the discharge side of the fan.

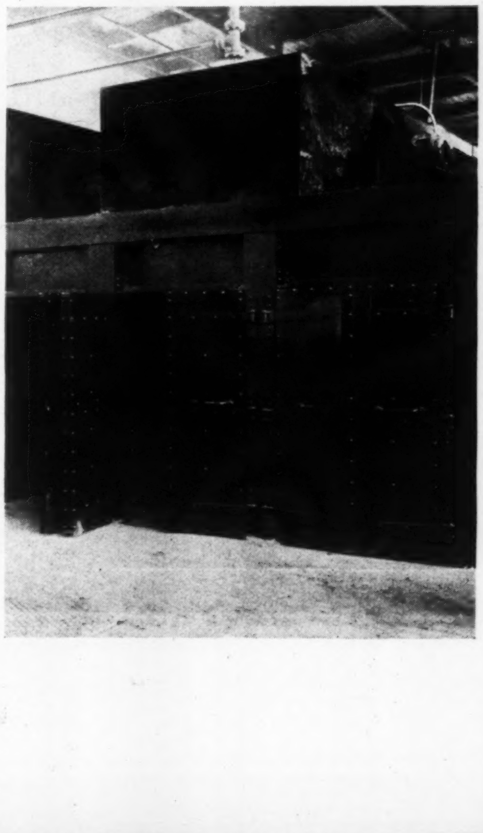
Drying Lime in a Wire Mill

Hot rolled rods have a coating of mill scale which must be removed by pickling before they are drawn into wire. Following this pickling process, the rods are lime coated (a wet process) to prevent rusting and to neutralize any trace of acid. During the cleaning process, the metal absorbs hydrogen. Unless removed by high temperature air, this will cause breaking of the rod in subsequent cold drawing.

In one wire mill, drying of the wet lime is accomplished by the equipment shown in the accompanying illustration, the drying action being the result of rapid circulation of air which, when above 212 F, absorbs a considerable amount of moisture. Heat transfer from the air to the material is accelerated by the rapid circulation. The material is then brought to the required temperature for the speedy removal of the hydrogen.

The apparatus used is constructed with insulated compartments, each equipped with its own heat element consisting of an enclosed steam coil and a fan so arranged as to recirculate the air over the steam coil and return it to the compartment. Various temperatures are possible for different grades of steel by regulating the steam pressure and the speed of the fan.

This method of drying and removing the hydrogen by heated air requires little attending labor and operating costs are confined to the cost of steam and power for the fan.



Apparatus in a Wire Mill for Drying the Wet Lime

Causes and Prevention of Corrosion in Refrigeration Piping

By J. A. Day

ONE OF THE serious problems confronting refrigeration engineers and those responsible for the continuous efficient operation of refrigeration machinery making use of a circulating brine is that of reducing corrosion. The corrosion of piping results in losses from the inefficient absorption of heat, stoppage or breakage of pipe lines, sticking of valves or other similar difficulties resulting in expensive shutdowns and large repair bills. Other forms of corrosion may prove equally costly. The refrigeration engineer is not alone in fighting this enemy. Any industry using metal piping or containers for liquids has its corrosion problem to solve before it can operate at its greatest efficiency.

During the last few years the problem of combating corrosion has received a great deal of attention from the chemical and physical standpoint. This technical investigation has yielded a large amount of valuable information that should be in the possession of every refrigeration or construction engineer. As this work has been carried on simultaneously by interested commercial enterprises and by open-minded technicians the results have been checked from several angles.

The results of these investigations indicate that the corrosion of refrigerating equipment is produced by two fundamental factors, (1) the oxygen content of the liquid in contact with the metal, (2) the reaction of the solution (its comparative acidity or alkalinity).

Discussing the first factor, it might be pointed out that water with a high content of dissolved oxygen is much more corrosive to metals than water that has been treated for the removal of the surplus oxygen. One investigator[1] studied the development of rust on polished steel through a binocular microscope magnifying 46 diameters. When a drop of water saturated with oxygen was placed upon the polished surface brown spots began to appear on the metal in 40 to 45 seconds. At the end of five minutes the rust was fairly well developed. When water saturated with carbon dioxide was used the rust developed around the edge of the drop in 60 to 80 seconds, while the remainder of the surface was not attacked for 6 to 8 minutes.

This worker concludes that electrolytic action begins in the presence of water according to the following equation:

$$Fe + 2H^{+}\ \text{(ionic)} + 2\ (OH)^{-} = Fe^{++}\ \text{(ionic)} + 2\ (OH)^{-} + H_2$$

Ferrous hydroxide is the product of this reaction and forms a film upon the metal, protecting it from furthur corrosion from this source. Carbon dioxide, however, reacts with this film, producing ferrous carbonate which is soluble, but the iron is reprecipitated in the form of ferric hydroxide when oxygen from the air dissolves in the solution as follows:

$$\underset{\text{Protective film}}{Fe(OH)_2} + \underset{\text{Dissolved CO}_2}{2\,H_2CO_3} = \underset{\text{Soluble}}{Fe(HCO_3)_2} + 2\,H_2O$$

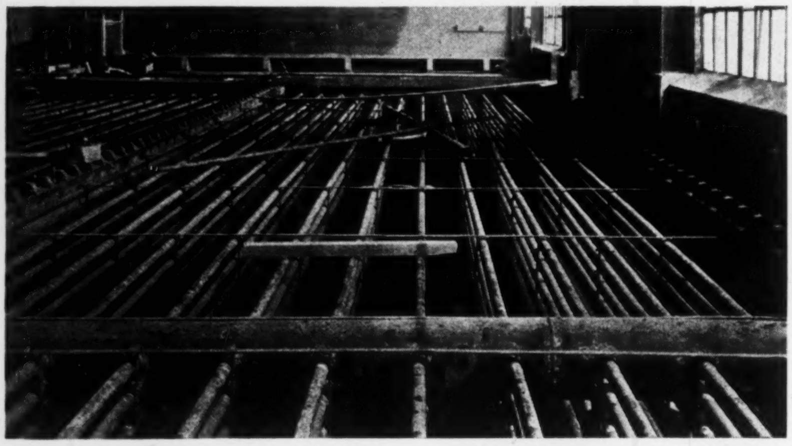
This work emphasizes the importance of dissolved oxygen as a cause of corrosion of ferrous metals. A weighed piece of polished steel will be found to lose weight when placed in corrosive solutions, the weight lost being in direct relation to the corrosive action of the solution. By this means the comparative corrosion factor of a solution may be determined.

Refrigeration Equipment Problems

Exposure of brine solutions to the air maintains a high oxygen content in the solution and aggravates corrosion. To avoid difficulty from this factor the brine circulating system should be kept as tightly closed as possible. In ice plants there is necessarily much exposure to air in the freezing tanks when the cans are being removed, but by efficient management these exposure periods may be cut down to the minimum.

The chemical reaction of the brine solution likely is the most important of the two factors and one that may be mostly easily regulated. Calcium chloride ($CaCl_2$) is most commonly used for making brine solution for refrigerating systems, although sodium chloride ($NaCl$) is still occasionally used. Brines made from these salts usually are very alkaline and the term "acid brine" is in reality almost a misnomer, although it is used to indicate a brine that is only slightly alkaline.

The zinc coating of ice cans and other equipment affords a protective coating against these corrosive factors so long as it is in its original condition. Chemical reactions take place, however, in which the metallic zinc combines with oxygen, moisture and carbon dioxide to form a covering of zinc oxide and zinc carbonate over the surface. When this stage is reached the degree of alkalinity of the brine becomes exceedingly important, as brines having a slightly alkaline or neutral reaction exert a much greater corrosive action than brines with a strongly alkaline reaction. Concentrated brines with specific gravities as high as 1.15 for sodium chloride and 1.20 for calcium chloride should be maintained. Under these conditions the coating of zinc carbonate is an addi-



Piping in an Ice-Making Plant for Supplying Refrigerator Cars tional protection, and if the brine solution is maintained at an alkalinity of 8.5-11.0 on the *pH* scale the corrosion will be greatly reduced. If new cans are first placed in a brine of low alkalinity a thin film of zinc hydrate will be formed, which is further acted upon by the air to form zinc carbonate, a gray, tough, protective coating. It is at this point that the alkalinity of the brine should be increased.

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[1] Jour. Industrial and Engineering Chemistry, Vol. 22, No. 6, 1930.

The introduction of the *pH* method (hydrogen ion concentration) of determining the relative acidity or alkalinity of solutions is especially adaptable in this problem. By the use of chemical indicators the *pH* value of a brine solution may accurately and immediately be determined. On the *pH* scale of 1 to 14.0, 7.0 is taken as the neutral point, all lower values representing acid conditions and all higher values representing conditions of alkalinity. Suitable test kits and indicators are on the market, so that making the test requires only the placing of a few drops of the indicator in a few drops of the brine solution and comparing the color formed with a standard color chart, which will give one directly the *pH* value of the solution.

REFRIGERATION SECTIONS IN A MILK COMPANY'S PLANT

Chemical Retarding Agents

Investigations along the line of retarding corrosion by chemical agents which might be added to the brine solution also have been fruitful, in spite of the fact that many chemical mixtures sold for this purpose have not given satisfactory results. The addition of neutral sodium chromate to calcium brine solutions at 100 to 150 pounds per 1,000 cubic feet has been reported as cutting the corrosion rate to one-tenth of the rate in untreated brine. The *pH* of such a solution should be between 7.0-9.0. With sodium brine 215 pounds of sodium chromate should be used per 1,000 cubic feet of solution.

Condenser Systems

The corrosion of the piping of condenser systems occurs chiefly from the dissolved oxygen contained in the water used. In re-circulating systems additions of small amounts of sodium chromate or sodium silicate to the water supply will effectively prevent the formation of rust and scale. In condenser systems employing fresh water that is used but once and wasted, the protection of pipes by painting or other surface treatment is the only recourse. A satisfactorily paint for this purpose, composed of lead chromate and red lead ground in linseed oil, has been developed as a result of researches conducted by the Paint Testing Committee of the *National Paint Manufacturers' Association.*

Based upon the results of the investigations reported above, correlated with good plant practices, the Corrosion Committee of the *American Society of Refrigerating Engineers* has presented the following recommendations:[2]

RECOMMENDATIONS BASED UPON EXPERIMENTAL WORK

BRINE SYSTEMS

Experimental work has indicated that the most satisfactory corrosion retarder for all brine systems is sodium dichromate. Due to the somewhat irritating and poisonous character of the dichromate when in contact with the body, an alternative treatment with disodium phosphate is suggested for open sodium brine systems. As a result of the number of plant applications which have been made with dichromate in open systems, the committee feels that with ordinary precautions there is little danger from chromates when the recommended concentrations of dichromate are not exceeded.

Calcium Brine Systems.—For calcium chloride brine it is recommended that approximately 100 pounds of sodium dichromate ($Na_2Cr_2O_7 \cdot 2H_2O$) per 1,000 cubic feet of brine be added (1.6 grams per liter). In addition there should be added sufficient sodium hydroxide to convert the dichromate to the neutral chromate. For neutral brines this amount is approximately 27 pounds per 100 pounds of dichromate; for ammoniacal brines the amount will, of course, be less and in some cases (when the brine shows red with phenolpthalein after the dichromate is added) none will be needed.

The dichromate may be hung in a bag in the brine at a point of rapid circulation or when convenient it is better to dissolve the retarder in a little warm water and pour the solution slowly into the circulating brine. The bag method has the advantage of requiring less labor and insuring better distribution. Approximately one-half of the original amount of dichromate should be added to old brine once a year; analysis of the brine for dichromate would, of course, be a more advisable way of determining the quantity required, but the amount specified will usually be satisfactory.

In making up a new tank of brine, it is recommended that the brine be mixed in a separate tank, dichromate (or chromate) added, with sufficient caustic soda to make the brine just alkaline to phenolphthalein and the whole be allowed to stand until the insoluble portion settles. After this is done, the clear brine may be pumped off into the new tank, cooled to the usual temperature and not until then should the cans be placed in the brine. By following this procedure the abnormally high initial corrosion of the galvanizing on the cans can be largely eliminated.

Calcium Magnesium Brine Systems.—For calcium magnesium chloride brines the same procedure recommended above should be followed except that 200 pounds of dichromate per 1,000 cubic feet of brine should be used. The method of making up the new brine (recommended above) is especially advisable in this case.

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[2] *Refrigeration Engineering.*

Sodium Brine System.—Sodium dichromate is recommended for sodium brine systems in concentration of 200 pounds per 1,000 cubic feet. The same procedure of adding alkali and renewing the dichromate semi-annually is necessary.

Phosphate.—For sodium brines in open systems the use of disodium phosphate ($Na_2HPO_4 \cdot 12H_2O$) is suggested as an alternative. Although not as effective as dichromate and requiring frequent renewal, it does not have the disadvantage of irritating and poisoning the skin of workmen. The proper concentration to use is approximately 100 pounds per 1,000 cubic feet of brine, and this amount should be added once each month. The brine must be kept neutral or only very slightly acid by the addition of muriatic acid if necessary. If phenolphthalein is used to test the brine and a pink color is observed, the acid should be added until the color just disappears. If the British Drug House universal indicator is used the color should be yellow. The retarder must be dissolved in hot water and added slowly to the brine at a point in front of the agitators.

CONDENSER SYSTEMS

Fresh Water Recirculating Cooling Systems.—The use of sodium silicate (water glass) is recommended for this type of system. No universal rule can be given for the amount to use, due to variations in local conditions. In general, 1.5 gallons of 40 deg. Be. silicate per 1,000 cubic feet of make-up water should be used at first. This amount should cause the water to show pink when tested with phenolphthalein after one hour, and if this coloration is not obtained at the end of one week's applications, the additions should be increased.

Sodium dichromate is also recommended for recirculating condenser systems, especially where the water comes in contact with corrosive industrial air. The amount required to stop corrosion varies considerably with the water and temperature and may be determined by immersing some clean steel in a continuously aerated sample of the treated water for a week. With the proper additions, only very slight corrosion should be perceptible in that time. The recommended concentration for first trial is 6 pounds per 1,000 cubic feet of make-up water. Caustic soda (1.75 pounds per 1,000 cubic feet) should also be added. Dichromate may be injurious to adjoining property when windage losses occur. It therefore has a rather narrow field of usefulness, although it is more effective than the silicate. A large excess above the amount required to stop corrosion should be avoided.

Non-Recirculating Cooling Systems.—Due to the large amounts of water used, no economical method has yet been devised for the treatment of non-recirculated fresh or salt water. Corrosion prevention must at present be obtained by use of rust-resisting metals or efficient protective coatings.

In all cases, threaded ends of pipes and hot-gas lines should be efficiently protected by the application of a good priming coat containing an inhibitor, covered by a good waterproof material. The surface of the metal should be perfectly dry and clean before painting or coating.

Cleaning Condensers.—It is recommended that as far as practicable condensers be kept free from slime and scale, especially during the summer months. The formation of a coating on the pipes raises power costs considerably, and in the average plant it seems safe to say that the condensers are cleaned far too seldom. Efficient water distribution is equally important in obtaining efficient heat transfer.

MISCELLANEOUS SUGGESTIONS FOR GOOD PLANT PRACTICE

BRINE SYSTEMS

1. In all systems steps should be taken to prevent, in so far as possible, the agitation of brine in contact with air; all inlets should be sealed. In general, the elimination of dissolved oxygen will largely prevent corrosion difficulties and any efforts in this direction will be well repaid.

2. It is suggested that weighed test pieces of galvanized and bare iron or steel be hung in the brine system as a convenient method of observing the degree of corrosion or protection resulting from treatment. Indications of pitting should be especially noted.

3. If brine is not treated it should be maintained slightly alkaline (about pH 8.5). The use of phenolphthalein as an indicator of alkalinity is recommended. A faint pink coloration should be maintained, if necessary, by the addition of a small amount of caustic soda or milk of lime.

4. Ammonia leaks should be repaired as soon as noticed. Sodium dichromate will usually give good protection to bare or galvanized iron even in strongly ammoniacal brine.

5. Contact of dissimilar metals should be avoided as far as practicable. The dichromate treatment will in most cases minimize this type of corrosion.

6. Strict cleanliness observed in and about the tank room in ice plants is advisable.

General.—1. When paint is to be applied for corrosion protection, it is especially important first to clean the surface thoroughly. Simply brushing off dirt is not sufficient. Removal of scale and dust is absolutely necessary, and heating to remove the small film moisture is advisable before the paint is applied. A good inhibitive priming coat followed by a waterproofing coat properly applied will often give good protection.

2. A suggested method of protecting exposed threads in galvanized pipe lines in cold storage is by the application of a lacquer, containing flake aluminum or aluminum bronze. All oil and dirt should be removed from the threads before painting.

Galvanic Corrosion

This form of corrosion is important to utilities, as in many communities the various utilities are owned by different interests and the matter of differentiating between soil corrosion and electrolysis frequently arises.

Galvanic corrosion has been reported to be greatly reduced or prevented entirely in some cases by electrolysis drainage, provided by connecting underground metallic structures to the negative busses of railway substations. This form of drainage permits the current to collect and counteract the discharged galvanic currents.

Problems in Rayon Manufacture

The rayon industry has one of the most serious corrosion problems. This industry uses large quantities of hydrochloric acid and sodium hypochlorite. Not only must storage facilities for large quantities of these solutions be provided, but they must be piped to different parts of the plant and conducted from place to place. The highly corrosive nature of both of these solutions has made the problem important. The entire equipment used in some plants is made from chemical stoneware that is not subject to corrosion from these chemicals.

Large stoneware storage tanks of up to more than 500 gallons capacity are used as reservoirs and the solutions are transported through pipes of the same material. These pipes are connected by means of armored flanges lined with stoneware that make tight fitting joints and yet permit no metal surface to come in contact with the solutions in the pipe. Valves lined with stoneware are used in the same way. This indicates the extent to which one infant industry has been forced to go to solve its corrosion problem which would determine its success or failure.

Handling Kitchen Exhaust in an Apartment House

By Samuel R. Lewis*

AN ENGINEER asks, "Why do I have so much trouble with the kitchen ventilation in my ten-story apartment building? We have a big exhaust fan in the attic pulling out of the vertical flue which serves the ten super-imposed kitchens, but we do not get proper results from it. Sometimes we smell onions cooking all over the building. Sometimes we find a good draft out through one of the openings and next day or next hour we find no draft, or a reverse current. I am puzzled—please help me out."

Unless we see that ten-story apartment house we must, of course, make some reservation in our diagnosis, but there is nothing in the history or symptoms which is not ordinary and regular, so that a shot-gun diagnosis probably is safe.

The engineer who designed the plant probably reasoned as follows:

"There are ten kitchens. We will give each one 300 c.f.m. exhaust, and will install in the attic a suction fan good for 3,000 c.f.m. at ⅝ in. static pressure."

The architect undoubtedly limited the engineer as to the area available for the vertical duct, which probably begins with an area of about 12 in. × 12 in., giving a velocity of about 300 linear feet per minute through the first story ceiling, and increasing in size as it comes up to about 18 in. × 24 in. through the tenth story ceiling.

Several "Ifs" to Proper Operation

Now if all ten kitchens were at the same temperature, and if each one had its window open the same amount, and if all the dampers in the branches were adjusted properly, and if all kitchens were on the same side of the building, and if part of them were not shielded from the wind by an adjoining building, *perhaps* each kitchen would lose 300 c.f.m. of air just as was planned.

But if some of the apartments are not rented and if the doors and windows to the kitchens are closed, and if the maid in one of the kitchens has the door to the back stairs open while she visits with the janitor, and if some other maid is house-cleaning and has her kitchen windows wide open looking to windward, we may expect peculiar and interesting developments from the ventilating system.

These are visualized to some extent in Fig. 1 drawing.

Effect of the Wind

A strong wind from the left may reverberate against the pent house and build up pressure to windward of it sufficient to throttle very materially the outlet from the exhaust fan. The wind whipping around the pent house door to the right acting as an ejector may create a very strong draft up the stair hall, which often has been proved to be stronger than the draft caused by the exhaust fan.

Thus, at the ninth story we see the exhaust opening reversed, spilling its odors from lower stories out into the kitchen and through the "unfortunately" opened door into the stair hall.

The tenth, eighth and fifth stories, with their windward windows open just a little may be receiving normal ventilation. The seventh and the sixth, and the fourth, have their windows closed and are hot and smelly, since there is no air supply to act as a carrier for the objectionable constituents.

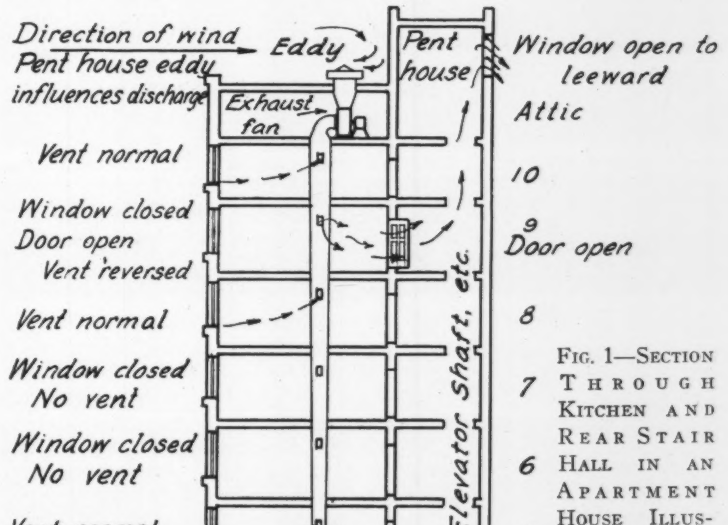
Direction of wind
Eddy
Pent house
Window open to leeward
Pent house eddy influences discharge
Exhaust fan
Attic
Vent normal
10
Window closed
Door open
Vent reversed
9
Door open
Vent normal
8
Window closed
No vent
7
Window closed
No vent
6
Vent normal
5
Window closed
No vent
4
Window open wide-Excess vent
3
Stair Hall, Elevator Shaft, etc.
Reverse vent
2
Door open
Adjacent building
Reverse vent
1

FIG. 1—SECTION THROUGH KITCHEN AND REAR STAIR HALL IN AN APARTMENT HOUSE ILLUSTRATING COMMON VENTILATION DIFFICULTIES

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* Consulting engineer, Chicago, Ill.

"It is of great importance in all lofty buildings that all doors into shafts which act like chimneys shall be weather-stripped, and that the walls of these shafts shall be air-tight."

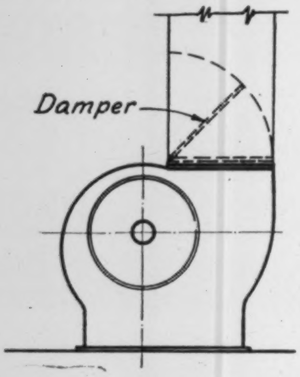
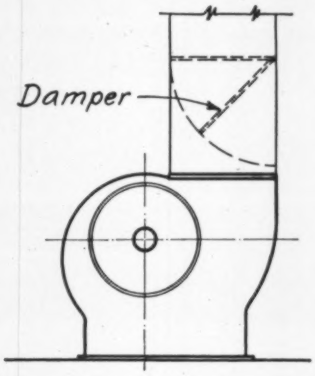


FIG. 2—INEFFECTIVE DAMPER ON A CENTRIFUGAL FAN

FIG. 3—AN EFFECTIVE DAMPER ON A CENTRIFUGAL FAN



The third has wide opened windows to windward and the great pressure thus created fills the full demand of the exhaust flue, so that the first and second story kitchens cannot get any normal exhaust ventilation whatever.

The second story window is open a crack, and the transom to the back stairs happens to be open. The room gets excellent ventilation, by supply down the erstwhile exhaust flue, and by exhaust into the stair hall, where everyone knows what the second story tenant will have for dinner.

When everyone is cooking, the air in the exhaust flue will be warm and light, and will tend to rise to the exhaust fan. When there is no cooking, the air in the flue may be quite cool, and will sometimes refuse entirely to rise, especially if there is an easily pulled warm supply of air from some higher-up kitchen nearer to the exhaust fan.

Air an Elastic Substance

All of this description will have failed unless we have enabled you to visualize an inflexible puller dragging at an exceedingly elastic substance which is so adaptable that it will change its weight very rapidly as it meets heat or absence of heat, and which, on meeting an obstruction, will hurry up temporarily, and will squeeze its volume by, through a narrow defile, only to slow down and to proceed as usual after passing it.

Did you ever see one of those big blood sucking leeches in the old swimming hole? They make an excellent illustration of this trick which air has of hurrying by an obstruction and afterward resuming its normal movement at reduced speed and enlarged area. The leech will perform the same trick as you pull him between your fingers.

Volume Control for Housed Fan

I remember long ago trying to reduce the cold stream of air being delivered by a centrifugal blower around some too-cool furnaces by damming up the fan-inlet.

I borrowed an old door and some planks and used my overcoat and my cap and thought surely I must have stopped the air delivery, only to find that just about as much cold air as ever was coming through the fan outlet, as the leaky obstruction at the fan inlet permitted many high-velocity streams of air to pass by.

I then built a damper in the fan outlet as shown in Fig. 2. There was enough weight tied to the damper to hold it in almost any position desired, but the method of throttling and regulating the volume of air delivered was unsatisfactory and irregular, due to the funnel-shaped restriction which encouraged the hurry-up, slide-by process.

I then learned that if I turned the damper around as in Fig. 3, so that the damper opened *against,* instead of with, the air stream, I could regulate the air volume exactly and consistently.

This method of volume-control for housed fans has many conveniences, especially as in many cases the power consumption, using it, will be reduced more nearly in proportion as the air volume handled is reduced than when we use resistances in the electric motor circuit which merely convert some of the electric energy to wasted heat.

Due to the constantly varying conditions of suction-pressure, there are very few, if indeed there are any, satisfactory schemes of multi-story kitchen ventilation by one overhead suction fan serving all of them.

A very few innocent looking little cracks in a long exhaust duct can prove exceedingly troublesome to the engineer who is trying to secure the actual volume of air delivery calculated, down at the far end.

Where we have the kitchen exhaust fan, as in many restaurants, churches, etc., in the kitchen itself, with one fan per kitchen, the fan pushing the air out by pressure through the long journey to and through the roof, we do not remember ever having had any trouble.

Wherever we have done the kitchen-exhausting by a long suction duct to a remote fan serving a number of kitchens, we have had trouble.

I know of one ventilating and air-cooling office building plant which has the supply fan in the basement and the exhaust fan in the attic. It works nicely until the heating season is over, when the interior air, even though about to be rejected, is cooler than the general air outside the building. We find then that we may as well stop the exhaust fans, since

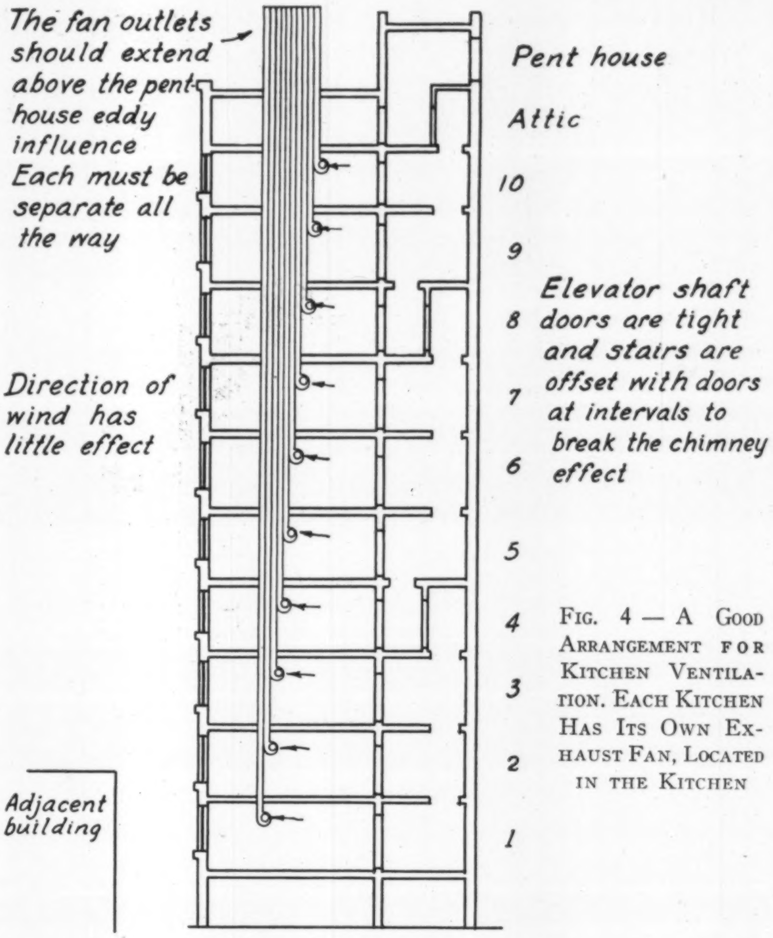

The fan outlets should extend above the pent-house eddy influence
Each must be separate all the way
Direction of wind has little effect
Adjacent building
Pent house
Attic
10
9
8
7
6
5
4
3
2
1
Elevator shaft doors are tight and stairs are offset with doors at intervals to break the chimney effect

Fig. 4 — A Good Arrangement for Kitchen Ventilation. Each Kitchen Has Its Own Exhaust Fan, Located in the Kitchen

they do not accomplish any useful work. They cannot lift the heavy cool air up out of the building.

This is why air-cooling ventilating plants in lofty offices and in big theaters have to recirculate.

Some day I will get one of these cooling installations in which I'll have the supply fan above the space it serves, with the exhaust fan *below* the space it serves. With this arrangement we ought to be able to move the air just about wherever we desire regardless of its relative weight.

Risers Must Be Tight

It must be remembered that either exhaust or supply risers used for kitchen ventilation or for bathroom ventilation, or even for fireplace ventilation in many-storied buildings, must be tight beyond peradventure.

The least leakage will upset conditions for a suction system, and may cause untold trouble by smell-leakage with a pressure system.

Individual Exhaust Fans for Kitchens

I believe that if each kitchen shall be given its own exhaust fan, in the kitchen itself, with its own separate flue all the way out of doors, we will have no trouble, and that only by such an arrangement can we be sure of no trouble. (See Fig. 4.)

A separate outlet flue for each fan is necessary, of course, as otherwise the fans which are running will cause back-draft into such kitchens as do not happen to have their fans running.

When it is not practicable to place a separate fan in each kitchen, the use of several attic exhaust fans, each coupled through separate risers to a group of kitchens, as shown in Fig. 5, will give far better results than will a single attic exhaust fan with its inevitable tendency to favor the near-by openings.

Drafts Complicate Problem

The situation always is complicated in tall apartment houses by elevator shaft draft, stair hall draft, etc. In some cases a tower with a ventilator or an open window or door will cause a greater suction than will the comparatively feeble exhaust fans on kitchens or bathrooms. The best remedy for this is in the architect's hands; to provide tight doors, fire barriers, etc., which will localize the building-suction.

It is of great importance in all lofty buildings that all doors into shafts which act like chimneys shall be weather-stripped, and that the walls of these shafts shall be air-tight.

Importance of Problem

Money spent in installing the proper ducts and fans, and in furnishing the right weather-stripping for doors is invested wisely, as the handling of kitchen exhaust so as to avoid cooking odors and faulty ventilation of the kitchens is of great importance. It is, of course, much more expensive to correct an improper installation after the building is finished than to install the equipment properly originally.

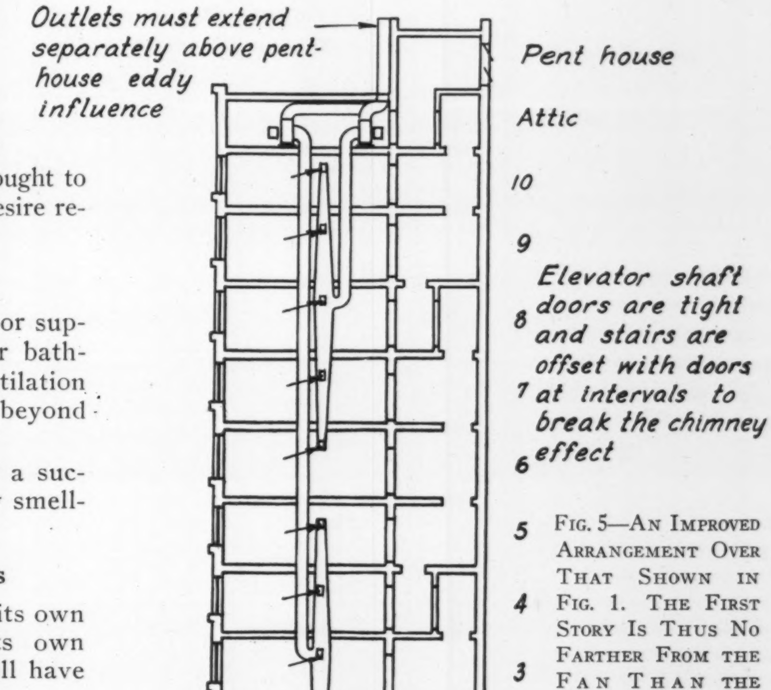

Outlets must extend separately above pent-house eddy influence
Pent house
Attic
10
9
8
7
6
5
4
3
2
1
Elevator shaft doors are tight and stairs are offset with doors at intervals to break the chimney effect
Adjacent building

Fig. 5—An Improved Arrangement Over That Shown in Fig. 1. The First Story Is Thus No Farther From the Fan Than the Fifth. The Fan Which Serves the Lower Zone May Run Faster Than That Which Serves the Upper

An Installation of Humidifying Equipment in a Card Room. The Units May Be Seen Attached to the Posts Above the Cards

Proper Placing of Humidifier Controls in Cotton Mill Card Rooms Important

By James W. Cox, Jr.*

PROVIDING proper, even and stable humidification in a cotton mill card room, in the opinion of many progressive textile mill men from the standpoint of quality and quantity of the sliver † produced, has been one of the most difficult humidification problems encountered in cotton mills. This has not been realized generally, however. The cause has been lack of exact information as to the amount of humidification really best in cotton mill card rooms; also lack of appreciation of how important proper humidification is in this department.

Until quite recently, a large majority of mill men considered that no humidification at all was necessary in a card room; a considerable proportion thought only a very low relative humidity was best, and a small proportion considered that some was needed perhaps, but it was not of great importance, whether it was high, low, constant or varied greatly.

The writer has found that most humidification troubles in cotton card rooms are due to lack of appreciation, or knowledge, of certain simple fundamental factors, and that when these are controlled by proper installations, immediate improvement in the whole situation as well as a change in the overseer's attitude become apparent.

A Common Method of Applying Humidity

The usual method of humidifying the card room in the average cotton mill (when it was humidified at all) has been to place a number of heads high up in the air and quite far apart throughout the room but in sufficient numbers to maintain in the upper strata of the room around the humidifiers, a relative humidity of between 50 per cent and 55 per cent at a temperature running between 75 and 80 F. The operation of these heads, in all but very large rooms, was regulated by one control box placed equally high up on a post in the center of the room. It was thought that this was all that was needed. In very large rooms there sometimes were two or more controls, but this was not a general practice.

Study Revealed Some Interesting Facts

A study of this matter in quite a number of mills, large and small, brought to light some very interesting facts which a great many superintendents and carders did not, at the time, consider to be of value, but which later proved to be of vital importance. The two main factors were: first, that the maintenance of semi-moist air, i. e. fairly high humidified air, in the middle or upper strata of the average card room and control of the relative humidity of this air close to the control instrument, or even around the heads, does not furnish sufficient moisture to properly humidify the cotton being processed; second, that the use of laps directly from the picker house, or after they have stood in the average card room for some time, is detrimental to high quality work.

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* Cox, Fuller & Mauersberger, consulting textile specialists, New York City.

† The product of the cotton or worsted carding machine. The same operation and product of the woolen carding machine is called "roping."

According to A. W. Thompson, air conditioning engineer, who read advance proof of this article, the importance of sound practice in the picker room as an adjunct to card room humidification cannot be too heavily stressed. Mr. Thompson has written a further discussion of some of the points mentioned in this article; his paper will appear in the "Open for Discussion" department in the December HEATING, PIPING AND AIR CONDITIONING.

As pointed out by Mr. Cox (and by Mr. Thompson) humidity regulators should be so located as to limit variations in humidity throughout the department as a whole to a commercially good basis of performance. With the proper installation, good operation will keep the system at its best efficiency.

Determination of the relative humidity of the air by sling or electric psychrometers at a considerable number of mills brought out certain interesting facts which varied very little among the large number of card rooms studied. These were:

1. There were three different strata of air in respect to moisture content.

2. At a level of about eight or nine feet, at which height the humidifiers and controls usually were placed, the relative humidity averaged about 55 to 60 per cent.

3. At a height of about six feet, i. e., the approximate height of the top of the cards, the relative humidity averaged about 50 per cent.

4. At a height of from two to three feet, at which the cotton both entered and left the cards and where it can be influenced most, the relative humidity averaged about 40 per cent (sometimes lower).

5. In this lowest strata, the lowest humidities were practically always nearest the doors or openings to the picker or other dry rooms.

6. The areas around the heads were higher in humidity than the areas between heads in all three strata.

7. The relative humidity near the control was higher than the average at equal height around the room, and lower than near the heads.

8. The relative humidity at the control was higher than in the corners of the room.

9. The relative humidity at the control, at the heads and other places was very much higher than at the openings or doorways leading to the picker house where almost always it was very low.

Determination of the moisture content of the stock in the laps just as they came into the card room, when on the cards, as well as in the sliver (which many mills pay no attention to whatsoever), brought to light further interesting facts. It was ascertained that:

1. The average moisture content in the laps was about 5½ per cent.

2. The variation in moisture content in the laps varied between 4½ and 11 per cent, even in short intervals of time.

3. The average moisture content of the laps usually was too low to expect well running work in the card room.

4. The variation in moisture content in the laps caused uneven moisture content in the sliver, hence uneven weight of sliver.

5. The variation, as well as low moisture content, lowered the quality of the carded sliver.

6. The average moisture content of the sliver was about 5½ per cent.

7. The lowest quality sliver was produced from fresh laps.‡

Generally speaking, the condition of the laps, sliver and air, in the many cases investigated, were almost invariably uneven and spotty. Proper moisture content was rarely in the stock. The proper conditions could not be obtained merely by raising the setting of the control to produce higher humidity.

A study of many such situations convinced the writer a long time ago that when the laps are of too low a moisture content, good operation of large humidifiers placed high up in the room can not be expected to overcome the average low amount of moisture contained in the laps. It convinced him further that the main thing to do was to provide a more constant amount of moisture in the laps before they were put on the cards; also that proper and even humidity had to be maintained at the level at which the laps entered the cards and at which the sliver was taken off the cards. It was decided that this was the only way to provide more even conditions, produce sliver of higher quality and lessen waste and cost of operation.

This, then, was a problem not solely of operation but of better installation and better operation later.

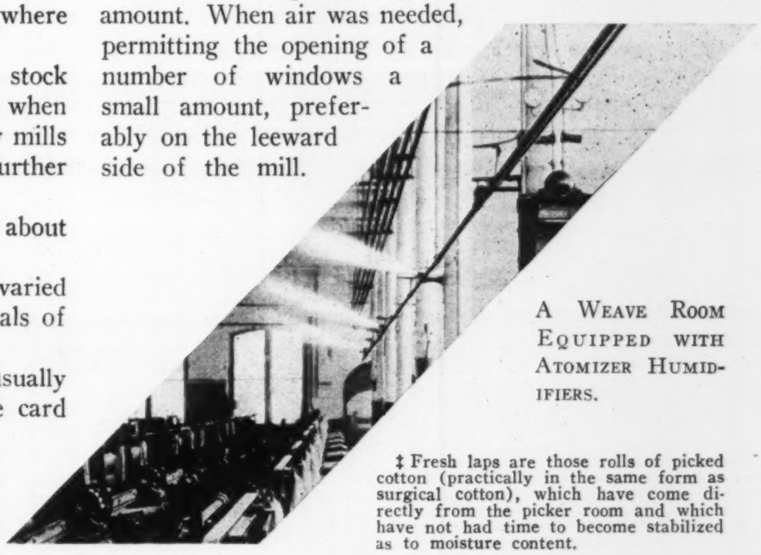
Improved conditions were obtained by:

A. Putting humidifiers at frequent intervals in the alleys and so placing them that the spray, to a certain extent, was forced downward towards the stock as it either entered or was delivered from the card.

B. Putting in more controls and setting them to give a higher humidity than usual, thus preventing to some extent the tendency to lose moisture in carding.

C. Closing of all openings, doors, etc. to the picker house, or any other connecting rooms which had normally a dry atmosphere.

D. Permitting no windows to be opened a large amount. When air was needed, permitting the opening of a number of windows a small amount, preferably on the leeward side of the mill.



A WEAVE ROOM EQUIPPED WITH ATOMIZER HUMIDIFIERS.

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‡ Fresh laps are those rolls of picked cotton (practically in the same form as surgical cotton), which have come directly from the picker room and which have not had time to become stabilized as to moisture content.

Complete Control of Air Conditions—Not Cooling Alone—Should Be Considered When

Air Conditioning Movie Theaters

By Malcolm Tomlinson*

ONE OF my most vidid recollections is of a Sunday morning in August at a motion picture theater. The weather outdoors was delightful. The air was neither cool nor warm. A little breeze was stirring the leaves of the trees. The sense of perfect comfort out of doors was deepened by the wonderful atmosphere within the building. A sacred music concert preceded the movie films so that the program was almost two hours in length. The music of the pipe organ, the play of the constantly changing color schemes and the thrilling plots of the pictures which followed came to an end all too soon. Reluctantly I arose to leave. As I passed from the lobby to the street a sudden hot blast shot through me. For an hour afterwards every pore of my body seemed to be busy throwing off perspiration. I am sure that I have never been so uncomfortable in all my life. I had come out of a cool and also pleasant atmosphere into a humid inferno. While I had been enjoying myself inside, the temperature and relative humidity outside had moved rapidly upward and the wind had died own. In the meantime the conditioned air of the theater had not been adjusted to meet changing weather conditions.

That one experience did more to make me weather conscious than all the weather reports and text books on climate which I had read. I had been a real movie fan. After that fatal day, I never entered a strange theater without considerable reluctance. Instinctively I always ask myself two questions. Will the inside atmosphere be comfortable and is the inside weather synchronized with that outside? You see I can't forget that wretched hour. It was entirely too high a price to pay for the two preceding hours.

There is nothing strange about this experience. Everyone has also suffered, and quite often it is real suffering, from the failure to provide a balance between the inside and the outside weather conditions. The time was, in the development of air conditioning equipment, that a successful installation was not expected to do more than furnish a fixed state of weather. That day has passed. There is no reason, today, why conditioned air should not be adjusted to meet any requirement as the need arises.

For theaters the comfort of the patrons is a real consideration. In extremes of weather, people go to the shows largely for the relief they hope to get from the outside weather. To keep this patronage, perfect comfort must not only be provided within the building but people must be able to leave with a grateful memory of that comfort. To obtain the last condition, the inside temperature and relative humidity must be patiently adjusted as the outside weather changes. Here is the crux of the situation. Theater management must not only install up-to-date equipment but must also be willing to operate it according to instructions. On the other hand, it is essential that the air conditioning industry shall do a thorough job of furnishing full operating instructions and even spend time training the operating engineers.

Cooling Alone Usually Stressed

There is one peculiarity about air conditioning in the theater industry. To the theater people, it has always been known as cooling. This is evident from the advertisements which are seen throughout the warm weather, and often over the whole year, on placards in front of the buildings. Here are a few samples of such appeals:

"Never Over 70 Degrees"
"Arctic Breezes"
"Cooled by Refrigeration"
"Cool and Comfortable"

The slogans are always accompanied by pictures of snow, icicles and blocks of ice.

No doubt there is a psychological effect, in hot summer weather when extremes of humidity prevail, through such advertisements. It is human to swing from one extreme to another and also to desire to do so. Yet to anyone who pauses to think the matter over, it would be evident that, even in warm weather, there are limits in cooling below which no human being could possibly be comfortable. In fact, I rather doubt that advertisements, such as the second and third in the list noted, actually help a great deal in keeping theater seats filled. "Arctic breezes" can only mean discomfort and the necessity for heavy clothing. "Cooled by refrigeration" is not likely to bring, to some minds, a more comfortable picture. No one stays in a cold storage plant by preference. In fact this latter ad is very likely to give people a picture of an unheated house.

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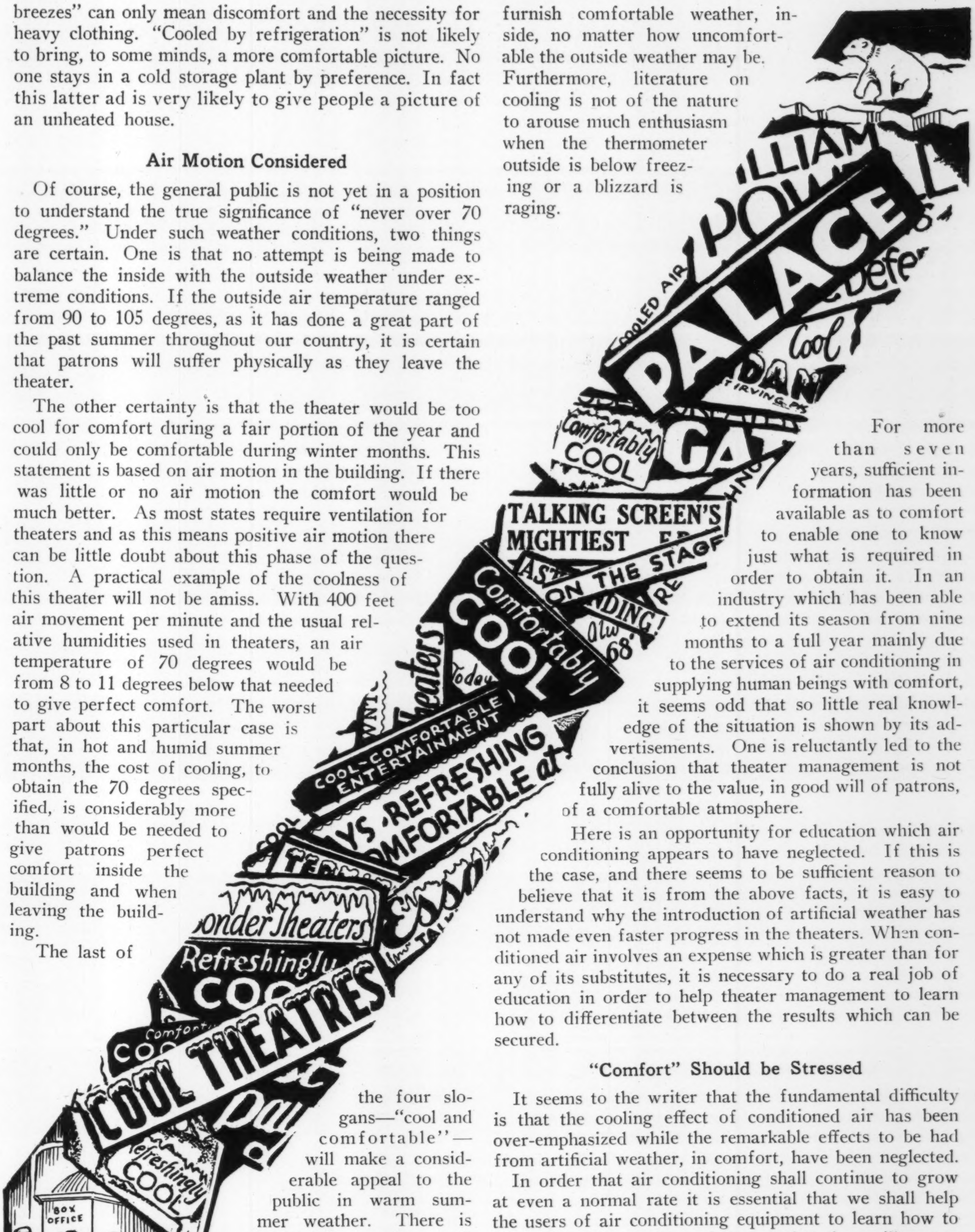
*Consulting engineer.

Air Motion Considered

Of course, the general public is not yet in a position to understand the true significance of "never over 70 degrees." Under such weather conditions, two things are certain. One is that no attempt is being made to balance the inside with the outside weather under extreme conditions. If the outside air temperature ranged from 90 to 105 degrees, as it has done a great part of the past summer throughout our country, it is certain that patrons will suffer physically as they leave the theater.

The other certainty is that the theater would be too cool for comfort during a fair portion of the year and could only be comfortable during winter months. This statement is based on air motion in the building. If there was little or no air motion the comfort would be much better. As most states require ventilation for theaters and as this means positive air motion there can be little doubt about this phase of the question. A practical example of the coolness of this theater will not be amiss. With 400 feet air movement per minute and the usual relative humidities used in theaters, an air temperature of 70 degrees would be from 8 to 11 degrees below that needed to give perfect comfort. The worst part about this particular case is that, in hot and humid summer months, the cost of cooling, to obtain the 70 degrees specified, is considerably more than would be needed to give patrons perfect comfort inside the building and when leaving the building.



The last of the four slogans—"cool and comfortable"—will make a considerable appeal to the public in warm summer weather. There is some doubt as to its worth through winter months.

Examine the advertisements of some of the theaters in your own neighborhood. Certainly there are few which advertise that they furnish comfortable weather, inside, no matter how uncomfortable the outside weather may be. Furthermore, literature on cooling is not of the nature to arouse much enthusiasm when the thermometer outside is below freezing or a blizzard is raging.

For more than seven years, sufficient information has been available as to comfort to enable one to know just what is required in order to obtain it. In an industry which has been able to extend its season from nine months to a full year mainly due to the services of air conditioning in supplying human beings with comfort, it seems odd that so little real knowledge of the situation is shown by its advertisements. One is reluctantly led to the conclusion that theater management is not fully alive to the value, in good will of patrons, of a comfortable atmosphere.

Here is an opportunity for education which air conditioning appears to have neglected. If this is the case, and there seems to be sufficient reason to believe that it is from the above facts, it is easy to understand why the introduction of artificial weather has not made even faster progress in the theaters. When conditioned air involves an expense which is greater than for any of its substitutes, it is necessary to do a real job of education in order to help theater management to learn how to differentiate between the results which can be secured.

"Comfort" Should be Stressed

It seems to the writer that the fundamental difficulty is that the cooling effect of conditioned air has been over-emphasized while the remarkable effects to be had from artificial weather, in comfort, have been neglected.

In order that air conditioning shall continue to grow at even a normal rate it is essential that we shall help the users of air conditioning equipment to learn how to use their apparatus in such a way that they will secure maximum results with a minimum of expense. As there are a very great number of combinations of temperature, air motion and relative humidity which will give perfect comfort we must do our utmost to show management how to pick, for each occasion, that particular combination which will give the best and most profitable results.

Comfort and weather go hand in hand. To obtain comfort, you have to provide artificial control of weather and this can not be accomplished in a satisfactory manner by failure to control that most vital of all weather factors—the relative humidity. This means humidification and dehumidification as the occasion demands. Each of these methods of air conditioning has its limitation but, together, they make a perfect whole.

Cooling equipment may be qualified to produce artificial weather. To do this it must, as we have seen, be able to control the relative humidity. Essentially a refrigeration plant, its special field is temperature lowering. As the temperature drops, a point is reached where the relative humidity has increased to 100 per cent. Further lowering of the temperature does not decrease the relative humidity. Therefore, to secure humidity control, heat must be introduced. By itself, then, refrigerating or cooling equipment is no more flexible than an ordinary house heating system. This, of course, is also true of humidification equipment. Without refrigeration or cooling, it is impossible to lower the air temperature more than a few degrees or to reduce the relative humidity to any considerable extent, in most cases.

It seems that the time has come when "Greenland's icy mountains," as the underlying theme of movie theater advertisements, should be discarded. The old order of things is changing. Comfort research has brought a new order into being. It is certain, today, that coolness does not always mean comfort any more than warmth means comfort. It is equally certain that cooling equipment, of itself, can not give perfect comfort throughout the year. Probably we need some new words with which to give full expression to comfort in all of its phases and also to simplify its story to the general public. At the present time, we air conditioning engineers have the term "effective temperature," which was coined by the research laboratory of the *American Society of Heating and Ventilating Engineers* to enable us to think of comfort in a single unit instead of three units. It seems doubtful whether the public will ever understand this term and it might be a very good thing if someone would invent a more expressive term for the same purpose.

It is not the purpose of this article to find the solution of the problems discussed but to bring these problems before the industry and profession. The theater needs the aid of the air conditioning engineer to sell itself to the public from a comfort standpoint. It's mighty hard to get a laugh, even out of Will Rogers and his wit, if you feel unbearably hot or cold. The movie and the legitimate stage need air conditioning in its highest sense. When they find it we will, in our turn, have better pictures and plays. It's a poor rule which doesn't work both ways.

Repairing an Air-Bound Drying Table

A manufacturer of asphalt paper used a steam table (See Fig. 1) for drying the paper after the asphalt had been applied and at least once a day the table became cold suddenly for no apparent reason and stayed that way for about an hour, after which it became hot again without any outside assistance.

The steam table was made out of four rectangular heating elements which were closed off on both ends. On the inlet sides, these ends were tapped in the center for 1-in. pipe and the outlet sides were just tapped in the lower section for a ¾-in. pipe. The piping connections were made as shown on the diagram except that the air discharge shown was not installed.

A ¾-in. high pressure thermostatic trap was taking care of the condensate from all sections and it was first assumed that the trap was not discharging regularly. However, this assumption proved false for even with a trapless discharge into the atmosphere, the table became cold.

It was then decided that air must be binding the table and preventing the heat from getting into the upper half of the heating element. The discharge side was then tapped ½-in., the table slightly pitched and a high pressure thermostatic trap screwed directly on the end of a horizontal run of pipe. However, that did not take care of the trouble either and only after adding a vertical run of six inches of pipe to the horizontal run of pipe and then connecting a high pressure thermostatic trap to that, the installation was foolproof.

Due to the very low pressure in the return main, it was possible to connect the return side of this trap to the return without spoiling results. If, due to leaking traps, the return main pressure would have been high, this connection would have made the installation troublesome again.—*E. M. Mittendorff.*

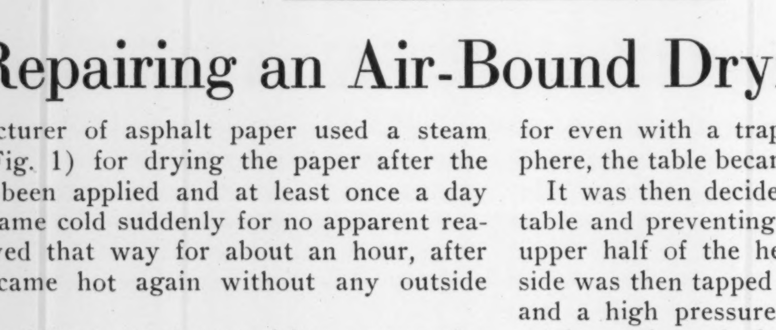


FIG. 1—A DRYING TABLE IN AN ASPHALT PAPER PLANT PIPED TO PREVENT AIR BINDING

"Open for Discussion"

A department in which we follow the custom of technical societies in allotting space in their programs for discussion of the papers presented

Expansion Bends

"Designing and Constructing Expansion Bends," *by G. W. Hauck.* Page 557, July, 1930 issue

THIS discussion is offered with respect to the article by G. W. Hauck, "Designing and Constructing Expansion Pipe Bends" which appeared in the July HEATING, PIPING AND AIR CONDITIONING. The method of computation used in this article apparently was based on data published in 1915; since then, much new information has been developed. Regarding this early method Mr. Hauck says, "Bends must be designed and the type selected for each particular problem or installation, and, while there exist exhaustive formulae, the exact basis of calculation can be readily obtained from the simple data following and given in table form."

The writer begs leave to take exception to the accuracy of the data on which Mr. Hauck's computations are made and to his statement that they afford an "exact basis of calculation." Such methods were discarded, to a great extent, years ago by engineers responsible for the design of major piping installations in this country and abroad, and the statement that they are "exact" should not pass without comment.

Prior to the time these early data were published, little information on this subject was available in English[1] and these data were therefore of material assistance to piping designers until more reliable methods were developed. The allowable amounts of expansion for different sizes and shapes of bends assigned by the investigators of 1915 were based primarily on empirical considerations deduced from a limited number of tests which they attempted to justify by checking against a formula intended for determining the strength of rings or crane hooks. Apparently the formula assigned to simple U bends was

$$\triangle = 0.0026\ R^2/r_0$$

where $\triangle$ is the expansion in inches required to produce 15,000 lb. per sq. in. fiber stress in the pipe wall assuming $E = 29{,}000{,}000$ lb. per sq. in.; R the mean radius of the bend; and r_0 the outer radius of the pipe. The corresponding formula for a simple U pipe bend generally accepted by other mathematicians and investigators is

$\triangle = 1.571\ FR^3/EI$, which on the same assumptions[2] reduces to $\triangle = 0.000813\ R^2/r_0$

or in other words, the expansive value of a simple U bend according to this formula would be 0.000813/0.0026 or 0.31 of the former. A further comparison along this line applying to several common shapes of bend will be found on page 551 of *Reference E* listed below.

No observations were made of the force required to produce a given displacement of a bend, the only data reported being displacements under load and permanent sets measured after release of load. Consequently no information was obtained regarding the corresponding reactions at anchor points. No attempt was made to set up a rational formula based on well established principles of engineering mechanics which would serve in applying test results from some particular bend to the diversity of pipe sizes and shapes encountered in actual practice. Due only to a lack of other information the allowable expansion values used by Mr. Hauck were used by many engineers responsible for the design of high temperature piping, and there was a feeling that the problem should be investigated more thoroughly and a logical basis of computation established. Several engineers[3] have tackled the problem since 1915 and advanced the solution step by step to a point where computed performance harmonizes with actual tests. The writer will agree that these later data are not as simple as those which they replace, but the results obtained are much more dependable for reasons which will be stated below. Fortunately the methods developed by recent investigators are in good agreement one with another, the principal need at the present time being to set these methods up in a form which is easier for the average designer to handle, or in other words to develop short cuts based on the rational methods. While considerable progress already has been made along this line, it is to be hoped that interested engineers will take up the problem and develop the practical application of these methods much further.

Exception is taken to the data and methods used by Mr. Hauck in the following respects:

(1) *Coefficient of Expansion*: The data on expan-

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[1] The following references were available in German:
(A) "Strains and Stresses in Expansion Pipe Bends," by Prof. A. Bantlin, Zeitschrift des Vereines Deutscher Ingenieure, Jan. 8, 1910, P. 43.

(B) "Regarding the Change in Shape of Thin Walled Tubes, Especially Expansion Pipe Bends," by Th. Karman, Zeitschrift des Vereines Deutscher Ingenieure, Bd. 55, 1911, P. 1889.

The following reference had been published in English at that time:
(C) "Elasticity and Endurance of Steam Pipes," by C. E. Stromeyer, Engineering (London, England), June 19, 1914, p. 857 (from a paper read before the Institution of Naval Architects).

[2] Both formulas as stated above apply to simple U bends with one end fixed and the other free to tilt during flexure. The effect of end restraint and of flattening of the circular cross section during flexure is neglected in both cases to put them on a comparable basis.

[3] Following is a partial list of references to supplement those given in the first footnote:

(D) "The Design of Pipe Bends for Expansion in Pipe Lines," by J. G. Stewart, Power, May 10, 1921, p. 742.

(E) "Elasticity of Pipe Bends," by Sabin Crocker and S. S. Sanford, A. S. M. E. Trans. 1922, p. 547.

(F) "The Elastic Deformation of Pipe Bends," by Prof. William Hovgaard, Journal of Mathematics and Physics (Mass. Inst. of Technology), Vol. 6, No. 2, Nov. 1926. See also subsequent papers by same author in that journal: Vol. 7, No. 3, 1928; Vol. 7, No. 4, 1928; Vol. 8, No. 4, 1929; also Proceedings of the National Academy of Sciences: Vol. 12, p. 365, 1926; Vol. 16, p. 444, 1930.

(G) "Stresses and Reactions in Expansion Pipe Bends," by A. M. Wahl, A. S. M. E. Trans., Vol. 50, No. 15, p. 241, Paper FSP-50-49, 1928.

(H) "The Flexibility of Plain Pipes," by J. R. Finniecome, The Engineer (London, England) Aug. 17, 1928 and following issues.

(I) "Design of Steam Piping to Care for Expansion," by W. H. Shipman, A. S. M. E. Trans., Fuels and Steam Power issue, Jan.-April 1929.

(J) "Piping Handbook" by J. H. Walker and Sabin Crocker, published by McGraw-Hill Book Company, Inc., 1930, Chapter VII.

sion of pipe given in Table 2 of Mr. Hauck's article show greater elongations per 100 ft. of pipe than is obtained from the formulas of Holborn and Day published in the *Smithsonian Physical Tables,* which probably is the most authentic information available. The following comparison indicates that actual expansions probably are considerably below the value of 6.89 for steel pipe obtained from his table.

Material	Linear expansion, inches per 100 feet 77 to 698 F.	Reference	Authority
Steel Pipe..	6.89	G. W. Hauck	None stated
Ingot Steel.	5.49	Marks' "Handbook"	Holborn and Day
Steel	5.31	Smithsonian Physical Tables 7th Rev. Ed., p. 218	Holborn and Day
Steel Pipe..	5.56	Special Test Trenton Channel Plant	The Detroit Edison Co., Research Dept.

The above tabulation gives the elongation per 100 ft. of pipe for a change in temperature from 77 to 698 F, as computed from coefficients of expansion obtained from the sources indicated, in comparison with the elongation measured in an actual pipe line.

A graphical comparison with Mr. Hauck's expansion values with the Holborn and Day data is given in Fig. 1 which is drawn with reference to an initial temperature of 70 F. While his expansion table agrees with data published in a number of piping supply catalogs, apparently all of them fail to state any authority for their values and are open to the same criticism. Further experimental investigation regarding expansion coefficients for pipe would seem to be in order.

(2) *Relative Expansion Values for Common Bends*: On page 557 of his article, Mr. Hauck assigns the following ratios for the expansive value of other common types of bends with reference to quarter bends: simple U bends, 2; expansion U bends, 4; double off-set expansion U bends, 5; circle bends, 5. Basing design on these ratios is unsound for three reasons: (a) the relative expansion values assigned by Mr. Hauck are incorrect (See *Reference E,* p. 551); (b) the blind use of such ratios is wrong since they vary according to how the bends are assembled with respect to adjacent piping, anchors, etc. (See *Reference J,* p. 563); (c) the coefficient K for flattening of the circular cross section of a curved pipe during flexure changes with the curvature and wall thickness of the pipe, thus upsetting the allowable expansion values assigned in his Table 2 (See *Reference J,* p. 550). Explanation of the above reasons would be too lengthy to attempt here, but is adequately covered in the references indicated. Incidentally, the allowable expansions which he assigns to expansion U bends and double offset expansion U bends are not as seriously in error as those assigned to quarter bends and simple U bends.

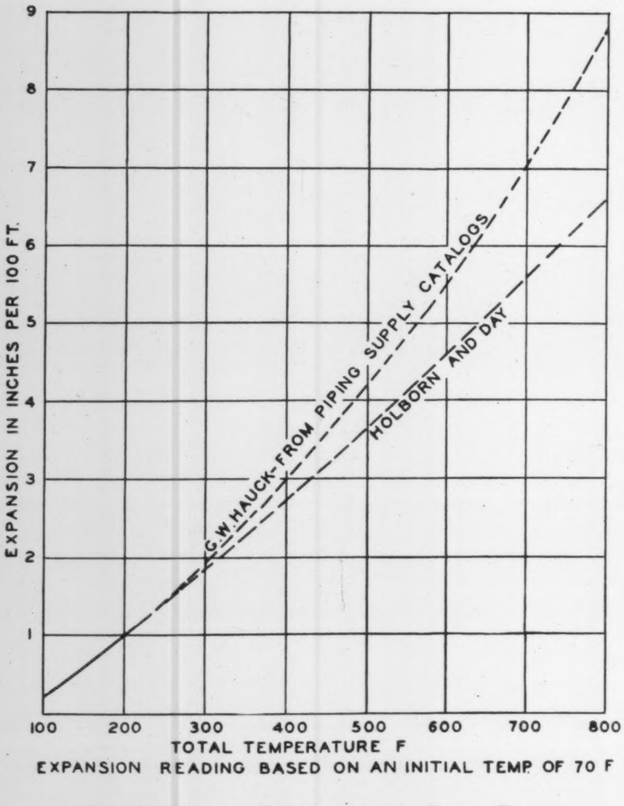
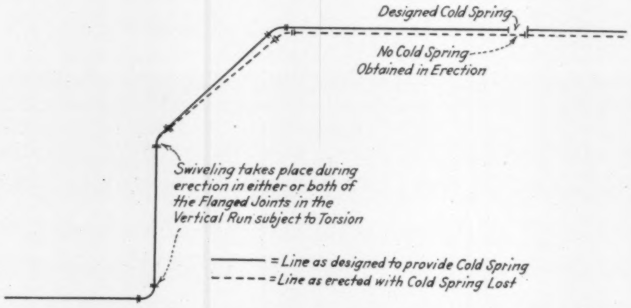


Fig. 1—Linear Expansion of Steel Pipe



Designed Cold Spring
No Cold Spring Obtained in Erection
Swiveling takes place during erection in either or both of the Flanged Joints in the Vertical Run subject to Torsion
= Line as designed to provide Cold Spring
= Line as erected with Cold Spring Lost

Fig. 2—Loss of Designed Cold Spring in Erection, Isometric View. (From Piping Handbook By Walker and Crocker, McGraw-Hill Book Co., Inc.)

(3) *Thrust at Anchor Points*: The methods used by Mr. Hauck take no account of reactions or thrusts set up at anchor points. Manufacturers of turbines and other equipment frequently specify the maximum allowable thrust against the connecting flange of their equipment. This is done to prevent damage to the equipment from excessive thrust, and safeguards the manufacturer's interests as well as those of the operators. It also is desirable to know the thrusts thrown against the building structure at intermediate anchor points in a pipe line. A method which does not furnish such data is inadequate for many applications.

(4) *Cold Springing:* Many pipe fabricators are prone to claim that allowable expansions can be doubled, or computed stresses cut in half, by cold-springing the line during erection of one-half of the expected expansion. This practice is questionable for several reasons. In the first place, if there are two or more runs of pipe at right angles to each other between anchors, it is possible for the line to swivel at the joints before the bolts are drawn up tightly, thus losing the desired cold-spring as shown in Fig. 2. Under such circumstances the flanges may come together solid without any cold-spring being applied, thus defeating the designer's intention of cutting the stress in half.

In case the desired cold-springing actually was realized during erection, it is apt to be lost in operation through piping taking a permanent set, especially in high temperature lines where more or less creep or plastic flow exists. It is a common experience with superheated steam lines to find that all traces of cold-spring have disappeared after a short period of operation. The more conservative practice is, therefore, to cold-spring the pipe on general principles and for the convenience it affords in erection, but not to count on this for reducing the stress or for doubling the allowable expansion. In other words, the line is designed as if no cold-springing existed.

(5) *Effect of Pipe Wall Thickness:* A statement is made that the expansion value of bends of corresponding shape made either of standard weight or extra strong pipe is the same. This is partially true as regards fiber stress in the pipe wall, but is entirely incorrect as regards thrust at anchor points. For a given shape and nominal size of pipe the reactions at anchor points are directly proportional to the moment of inertia of the pipe which in turn is a function of the wall thickness.

Take for instance the case of 6-in. pipe used in his example of a 6-in., 600-ft. steam line. Six-inch standard weight pipe has a moment of inertia of 28.14, while 6-in. extra strong pipe has a moment of inertia of 40.49. In this case the reaction at the anchor points with extra strong pipe, neglecting the effect of the flattening of the cross-section of the curved pipe, would be 40.49/28.14 or 1.4 times that obtaining with standard weight pipe.

Wall thickness also affects factor K used to compensate for flattening of the circular cross-section of a curved pipe during flexure. Referring again to the same example, the K value for 6-in. standard weight pipe bend having a mean radius of curvature of 50 in. is 0.732, while the K value for the same bend made of extra strong pipe is 0.875.

Due to the fact that the outermost fibers of the pipe tend to relieve themselves of stress through flattening of the circular cross section during flexure, there is a lessening in the resistance offered by the bend to flexure which is equivalent to a reduction in the moment of inertia, I, of the pipe section. Taking the above K values into account, the reaction at the anchor points with extra strong pipe is found to be (see appendix) $6{,}100 \div 3{,}630 = 1.7$ times that obtaining with standard weight pipe.

In case Mr. Hauck had used a shorter radius bend in his example, the discrepancy would have been even more pronounced. To give an idea of the numerical significance of the point, this example has been recomputed by the present writer using rational methods and the results presented in an appendix to this discussion.

Assuming that pipe wall thickness has no effect on the expansion value of a bend leads to a similar error in computing stresses, since the K value enters into this relation also. The net effect is not so far wrong as with the reacting force explained above, since the effect of change in I cancels out in the case of stress, leaving the change in K as the principal disturbing factor.

(6) *Straight Pipe:* No attempt was made to take into account the flexural possibility of runs of straight pipe at right angles to each other or in "square loops." The elastic properties of straight pipe are even more important than those of bends in most piping layouts. (See *Reference J,* pp. 517-570.)

(7) *End Connections:* No account is taken of the effect of end connections or the location of guides or anchors on the flexural properties of bends. It is uncertain whether the tests from which the data used by Mr. Hauck were obtained were made on bends with one end free to tilt during deflection, which is equivalent to testing with both ends supported on knife edges, or whether both ends were partially restrained. Bends installed in actual pipe lines have both ends more or less restrained from tilting by attachment to the adjacent piping. Computation of the elastic properties of piping should take into account the effect of end attachment by including all pipe between anchors in the flexural computations. (See *Reference J,* pp. 517-570.) A convenient means of taking into account points 3 to 7, inclusive, is shown in Fig. 3.[4] This figure applies to the use of expansion U bends under circumstances

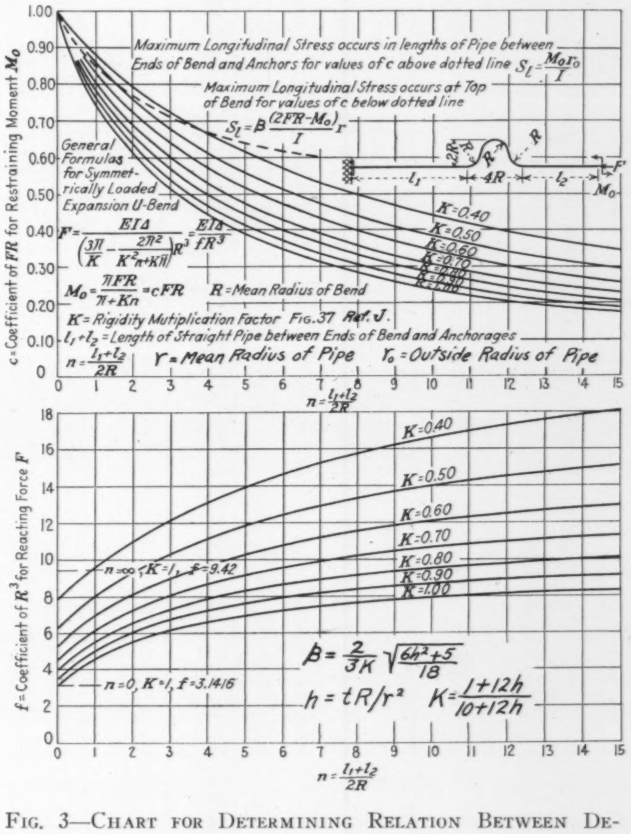


FIG. 3—CHART FOR DETERMINING RELATION BETWEEN DEFLECTION AND REACTING FORCE FOR AN EXPANSION U-BEND INSERTED IN A RUN OF STRAIGHT PIPE HAVING FIXED ANCHORS AT THE ENDS. ALL DIMENSIONS IN INCHES. FOR FURTHER EXPLANATION SEE REFERENCE J, PP. 550-568

equivalent to those assumed in the 6-in., 600-ft. pipe line of his example.

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[4] Fig. 3 was taken from page 565 of *Reference J* and modified to take advantage of certain simplifications in combining stresses pointed out by Professor William Hovgaard in his paper, "Tests on High Pressure Pipe Bends," Journal of Mathematics and Physics (Mass. Inst. of Tech.) Vol. VIII, No. 4, Dec. 1929. According to Prof. Hovgaard's analysis the *maximum longitudinal* stress at the middle surface of the pipe wall as determined from Fig. 3 can be used as the practical basis of pipe bend design, disregarding the effect of *transverse* bending stresses incidental to flattening of the circular cross section. If the *transverse* bending stress at the top of a bend is combined with the *longitudinal* stress at that point, excessively high values are obtained which apply only in a very localized layer of material at the outer edge of the pipe wall. This high local stress may be considered as relieved by plastic flow and disregarded in the computations. According to Professor Hovgaard, satisfactory results will be obtained provided the *longitudinal* working stress at the middle surface of the pipe wall does not exceed 16,000 lb. per square inch. This working stress should be considered as the total combined stress including the *longitudinal* stress due to internal fluid pressure minus any direct compression due to expansion thrust at anchor points. Where *torsional* stress is present it should be combined with the total *longitudinal* stress and the direct *radial* stress, due to the internal fluid pressure, as shown on pages 557 to 560 of *Reference J.* Here again the localized *transverse* stress incidental to flattening of the circular cross section may be neglected.

(8) *Change in Modulus of Elasticity:* No mention is made of the change in modulus of elasticity with temperature. The curves of Fig. 4, which are plotted from a formula developed by G. A. Orrok, show a decided decrease in the modulus with increase in temperature. This decrease in E tends to increase the flexibility of a bend at elevated temperatures and should be taken into account in the computations.

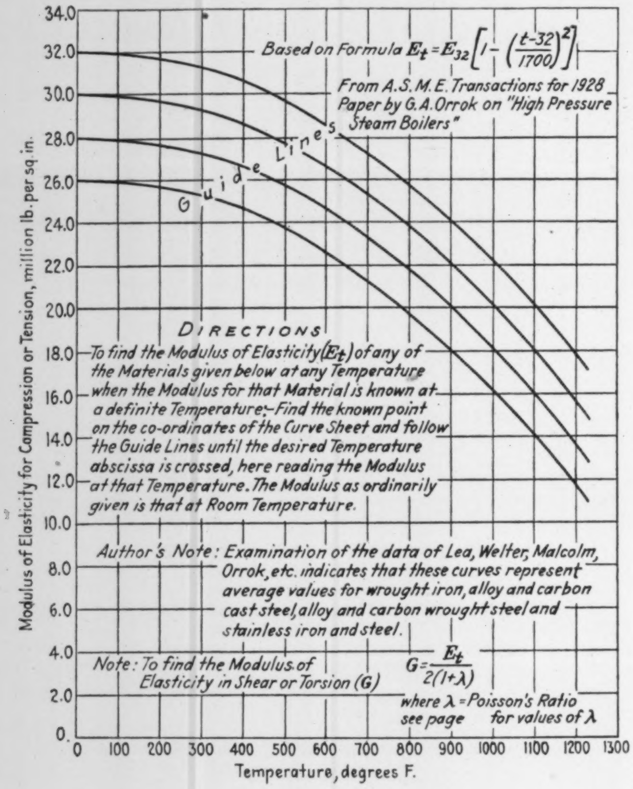

Fig. 4—Change in Modulus of Elasticity With Temperature. (From Piping Handbook, By Walker and Crocker, McGraw-Hill Book Co., Inc.)

(9) *Bending Moment at Joints:* It is impossible to determine the bending moments at pipe joints by Mr. Hauck's method. The amount of bending moment obtaining at any joint has a decided bearing on the probability of keeping that joint tight. Joints subject to high bending moments are much more apt to leak than those where low bending moments obtain.

A design in which the bending moments are pretty well equalized at all joints throughout the line is better in this respect than a design where a few joints are subject to excessive bending moments and the balance are not loaded anywhere near capacity.

A comparison of Mr. Hauck's results for his 6-in., 600-ft. pipe line example with those obtained by the method of Fig. 3 is furnished in the appendix to this discussion.

The above points are mentioned to call attention to the necessity for taking into account all factors concerned. Exact methods are fully as necessary in the design of piping as they are in the design of bridges, structural steel or machine parts. The methods used by Mr. Hauck are too inaccurate for most modern piping work and should be discarded, the writer believes. It is well worth any piping designer's time to read the references listed in connection with this discussion.—*Sabin Crocker,* Engineer, The Detroit Edison Co.

Appendix

Comparison of Results for 6-In., 600-Ft. Pipe Line: Several of the points discussed above can be brought out to best advantage by using rational methods for recomputing Mr. Hauck's example of a 6-in. steel pipe line 600 ft. in length, carrying 225 lb. saturated steam (temperature 397 F) in outdoor service, where the minimum temperature is figured at zero. Mr. Hauck's figures are repeated in parallel with the writer's for convenience in reference.

Solution:

Elongation in 200 ft. (0 to 400 F)
Mr. Hauck's article..................6.90 inches
Holborn and Day formula............6.46 inches
Modulus of elasticity for steel pipe at 400 F
Mr. Hauck's article......30,000,000 lb. per sq. in.*
G. A. Orrok data........28,600,000 lb. per sq. in.

For convenience in solution the writer recommends in this case the special application method shown in Fig. 3. This affords a simplified but rational solution for the case of an expansion U bend having runs of straight pipe on both sides of the bend, with anchors at the ends remote from the bend. As stated in a preceding paragraph, the K value for this bend if standard weight pipe is used is 0.732, while for extra strong pipe it is 0.875. Using this method the following values are obtained in comparison with the original solution. (Stress due to steam pressure is not included.)

Maximum longitudinal stress in bend for standard weight pipe:
Mr. Hauck, based on 15,000 × 6.9/8 and taking credit for cold-spring = 13,000 lb. per sq. in.
Mr. Hauck, based on 15,000 × 6.9/8 without credit for cold-spring = 26,000 lb. per sq. in.
Method of *Reference J,* page 565 = 34,500 lb. per sq. in.

Maximum longitudinal stress in bend for extra strong pipe:
Mr. Hauck, based on 15,000 × 6.9/8 and taking credit for cold-spring = 13,000 lb. per sq. in.
Mr. Hauck based on 15,000 × 6.9/8 without credit for cold-spring = 26,000 lb. per sq. in.
Method of *Reference J,* p. 565 = 46,500 lb. per sq. in.

Reacting force at anchors, standard weight pipe:
Mr. Hauck—not obtainable.
Method of *Reference J,* p. 565 = 3,630 lb.

Reacting force at anchors, extra strong pipe:
Mr. Hauck—not obtainable.
Method of *Reference J,* p. 565 = 6,100 lb.

Bending moment at pipe joints, standard weight pipe:
Mr. Hauck—not obtainable.
Method of *Reference J,* p. 565 = 29,600 in. lb.

Bending moment at pipe joints, extra strong pipe:
Mr. Hauck—not obtainable.
Method of *Reference J,* p. 565 = 42,800 in. lb.

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* If this value of E is placed at 29,000,000 rather than 30,000,000 the value of 28,600,000 should be proportionally reduced.

Improved Nomenclature

"Suggested Reforms in Heating and Air Conditioning Nomenclature," *by Samuel R. Lewis,* Page 744, September, 1930, issue

AFTER reading, with much interest, the article on nomenclature by Samuel R. Lewis in the September number of HEATING, PIPING AND AIR CONDITIONING, I venture to respond to his implied invitation to comment and make suggestions. This is from the point of view of an engineer in an architect's office.

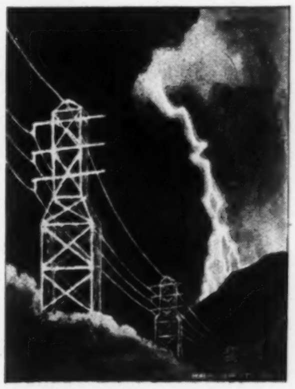
Surely the time is at hand to relieve the old horse as a standard for power measurement; but it seems to me desirable to retain a mechanical measure as distinguished from an electrical one. This might well, however, be the mechanical equivalent for the kilowatt. I suggest the term "unipower" for this new standard unit of power.

The name "unit heater" seems to me a clear and specific term for a definite piece of heating apparatus. It is in very general use at the present time; and I would like to see it retained. Heating unit, as a general term, should not confuse anyone. "Unit ventilator" also seems quite definite and satisfactory; and I would extend the designation to include "unit conditioner" to denote an assembled apparatus for cleansing and moistening the air in addition to tempering it.

For the convection heater, however, I think the terms *fluecon* and *fancon* meet a real need. Most of us have just stopped trying to make any designation fit; and have specified what we wanted by trade name, especially in the case of the *fancon.* The need for uniformity in measuring the *head* of a *fluecon* is quite apparent; and it would seem as though the *A. S. H. V. E. Guide* might take a hand to designate this. And if this were followed up in the tables in the advertising section of the *Guide,* it would establish a standard very quickly. I suggest, however, the term "heat-head" as the word draft is generally associated wtih the air required for combustion.

The remaining terms and definitions that Mr. Lewis has suggested are certainly well considered and worthy of standardization in the profession; and it would seem as though they might be included in the next *Guide* without fear of criticism, and as a progressive step in unifying our terminology.

And last, but not least, there is our old friend "square foot of radiation" and his small but pretentious brother "British thermal unit." To be sure, the latter is usually known by his initials, but even that is too much of a name for so small a unit, and one seldom used with precision except in the laboratory. I think someone, a while ago, suggested in one of our magazines the term "K-B" as a unit of 1,000 Btu for figuring radiation and other heat quantities; and it appealed to me at the time as a step in the right direction. My own suggestion would be a unit equivalent to 100 Btu and called a *therm;* with the possibility of using *centitherm* or *kilotherm* for a larger unit comparable to that other mis-measure of old Dobbin's perspiration that we designate a boiler horsepower.—*Edward B. Lewis.*



Use of "Therm"

I do not agree that unit heater and unit ventilator are satisfactory terms for what I mean when I say *elecon* and *elevent.* I do concur with Mr. Edward Lewis in his other statement. The therm should come into general use, and I am for centitherm and kilotherm. There seems no reason either for repeating the capital "B" before heat unit in ordinary practice.—*Samuel R. Lewis,* Consulting Engineer.

"Suggested Reforms in Heating and Air Conditioning Nomenclature," *by Samuel R. Lewis.* Page 744, September, 1930, issue

I HAVE been much interested in the article entitled "Suggested Reforms in Heating and Air Conditioning Nomenclature," and I certainly think some revision is desirable.

This matter of specifying heat units is particularly troublesome. It is bad enough in steam heating but I find practically all heating engineers are thinking of heat in terms of steam and it is doubly hard to make them realize that electric heating, which I principally deal with, simply can not be referred to in "square feet."

There is a marked difference between steam radiators and electric heaters, owing to the fact that the former are inherently constant temperature surfaces (for a given steam pressure) and give off variable quantities of heat according to the facility with which the heat can be carried away; whereas, electric heaters are constant energy input devices that will insist on delivering the goods or die in the attempt.

In other words, the temperature will rise to a point at which the given amount of heat will be delivered or the heater will destroy itself.

Anyone interested in figuring electric heating should follow the general tables for computing heat losses as published in the *Guide* of the *American Society of Heating and Ventilating Engineers,* and then convert the total requirements of Btu's per hour into kilowatts per hour, by simply dividing the total Btu's by 3,415, as 1 kilowatt hour is equivalent to 3,415 Btu's per hour.

For my own convenience, I have converted the tables for many of the standard types of construction so that they will read directly in watts per square foot instead of Btu's per square foot, and I find this a considerable time saver.

While electric heating is free of some of the problems of steam heating, it has a whole flock of its own. One thing, however, which would make life easier for the electric heating engineer woud be more accurate specifications of heat requirements, along the line advocated in this article.—*Lee P. Hynes,* electric heat engineer.

INTERVIEWS OF INTEREST

University of Chicago Air Conditions Greenhouse

During the period of six years just past the University of Chicago has carried on an extension program which has involved expenditures of $20,000,000 for new building. At the present moment many projects are developing. A new orthopedic institution is nearly complete. Chicago Lying-In Hospital is about three-fourths complete, and Oriental Institute is about half finished. A men's residence hall is under way. Bids are being received on International House and a graduate building. Plans are being prepared for a women's residence hall and for a new field house, while studies are being made for a new art building.

A large percentage of this money is paid for piping. It is not surprising therefore to find advanced methods and design that give unusual economies. Bernard Albert Eckhart Hall has the unusual provision of oxygen piped to all its physics research laboratories. Its block tin piping for distilled and redistilled water supply is interesting.

New Steam Plant

"It is the University of Chicago's new steam plant, however, which commands chief present interest on the part of outside engineers," states Lyman R. Flook, superintendent of construction for the University of Chicago. "The distribution system serves a volume of some sixty million cubic feet, steam being generated at 275 pounds pressure, 150 F superheat. The old electric plant has been shut down. Henceforth, electrical current will be purchased.

"The power plant is located on the Illinois Central tracks, convenient for transportation facilities. Power distribution is made by means of a new steam transmission tunnel 8 feet wide by 7 feet high and 5,000 feet long. Hangar-beams, inserts and piers support transmission lines. Expansion provisions are made at intervals of 300 feet, with fresh air intakes at each expansion area. The tunnel roof slab ranges between 4 and 10 feet below surface levels, being partly below water.

Metering Important

"The cost of steam is high. The uses made of steam became our proper study. Fifteen separate buildings on the campus now are metered, each with a different set-up, so that when final operation routines are developed each conservation measure can relate to specific equipment data. One building has full thermostatic control. One is unit control. One has a dual system. Another is controlled by zones. One tests out certain insulation. One will give facts on weather stripping. We mean to prove out operation changes thoroughly.

Conditioned Air for Seed Study

"A great university," states Mr. Flook, "forms partnerships with engineers on every hand. The layout of the new Botanical Building just opened especially reflects these relationships. We condition air for the study of seeds. A cold room 9 ft. 6 in. × 12 ft. 2 in. × 6 ft. 6 in. has been arranged in the botanical laboratory basement in which an unvary-

AIR CONDITIONING CABINET USED IN BOTANY GREENHOUSE AT UNIVERSITY OF CHICAGO ing temperature of 40 F below zero can be maintained. It has cemented floor and walls and ceiling, lined with 12-in. cork. Two other cold rooms are held constant at 15 and 32 F. These have dimensions of 10 ft. × 13 ft. × 6 ft. 9 in. each.

"Each of these cold rooms in turn is supplied with separate cabinets electrically equipped and insulated throughout with 4-in. cork to be individually operated at any temperature required with precise control.

"Air conditioning for plant life calls for many variations. One room is treated with fine spray to precipitate all dust. Live steam in some of the green houses is piped to soil beds to kill all bacteria. Potted dirt is treated in self-contained sterilization units.

"Some units here are not seen elsewhere. They operate within the several greenhouses and obviate the difficulty of trying to maintain constant conditions in large glass areas exposed to sunshine. These 'rooms within a room' are made of glass, some ordinary glass, some permeable to violet rays, and some with vitaglass only on the top. Sliding side openings permit easy handling of the plants.

Simulate Tropics

"These cabinets are capable of simulating tropical growth conditions. The basic principle of the cabinets is the use of a small central station air conditioning unit with fans, steam heating coils, controlled humidification, and closed circuits. Control arrangements contemplate holding conditions constant over periods long enough to measure growth changes incident to atmospheric variations from germination to seed bearing, the entire life cycle of the plants observed."

Building Power Plant Records of Value

Efficient management routines and accounting methods that budget time and materials as well as money costs and reflect at once an influence on work schedules have been worked out by M. Ehrlich, engineer of the Lake-Michigan building, Chicago. Tenants within this building consistently report the situation as exceptionally well handled from the occupant's viewpoint. Management is efficient, state the owners. Evidence like this justifies some effort to find out what goes on in the engine room.

The Lake-Michigan building is of reinforced concrete construction 23 stories high. (Alfred Alschuler, architect). This includes penthouses on the roof. Windows are all steel, double hung sash; weather stripping for windows with northwest exposure was part of the original building contract. There are seven elevators with variable voltage control.

There are three boilers, each with a load of 1,750 square feet of radiation. Cast iron radiators are used. Heating is by means of a two-pipe vacuum return system. The boilers maintain 90 pounds pressure. Return is 12-15 inch vacuum with return temperature maintained between 145-150 F. Even at 10 deg. below zero, the plant never carries more than a 2-in. vacuum on the supply side.

Mechanical arrangements for ventilation are confined to the stores on the first floor. One supply fan, 12,000 c.f.m. and one exhaust fan 7,000 c.f.m., motor driven, take care of this system. Oil-type filters, manually cleaned, are used and ventilation air is introduced at 75 F.

The hot water supply is a circulating system from a 12,000 gallon tank heated by steam coils in a closed circuit. Cold water supply units are in all the rooms. The tenants provide any electric water coolers that are used. The building owners ice all iced coolers. Compressed air can be supplied to every suite. From one to ten individual lavatories are allotted to each tenant.

Building Management a Double Problem

"There is a double problem in building management," states Mr. Ehrlich. "One must contrive somehow to maintain absolute service in the face of a great many variables; and one must produce this service at the lowest possible cost.

"Management routines, then, must be recorded in a manner that reflects precisely what is going on within the building. Our power plant comparative statement, first, sets out money costs; for salaries, for fuel, for ash removal, for equipment maintenance, for supplies.

"A second division of our comparative statement then undertakes to express this money outlay in terms of what it purchased. Water evaporation per pound of fuel consumed; the cost per 1,000 pounds of steam based on fuel expenditure plus ash removal; the steam consumption by areas within the building; the steam used per square foot of heating surface; these are some of the factors to be traced in management routines.

"A third expense accounting distributes costs by mileage and kw. consumption of elevators, and water used in the building.

Overheating Expensive

"Overheating is expensive. Our routines are worked out with full appreciation of the fact that the office building occupant will open windows when too warm. He will not shut off the heat. Overheat, then, is wasteful and every effort should be directed to keep the rooms continuously in the comfort range.

"We produce our steam as economically as possible. This means a special study with full test records of many kinds of coal.

"We weigh all coal we burn. We weigh all water supplied to boilers. Each man reports for his own shift on coal used and water evaporated, with corresponding CO_2 records and stack temperature readings.

"These operation records are assembled every twenty-four hours. This assures that no failure of management or equipment can go unnoted for long. When any lapse occurs, we know which operative was in charge. This encourages a competitive effort on the part of the men that has improved our showing. Two shifts are operated during the summer, but three shifts are used in winter months.

Lake-Michigan Building. Power Plant Comparative Statement. Year Ending December 31, 1930

	Jan.	Feb.	Mar.	April	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.	Totals
Salaries Paid $	977.32	899.16	862.68	725.48	738.56	760.80	717.60	724.00	703.20				
Salaries Applied To Power Plant 51% of Total $	498.43	458.57	439.97	369.99	376.66	388.00	365.97	369.24	358.63				
Salaries Applied To Building Equipment Maint. 49% of Total $	478.89	440.59	422.71	355.49	361.90	372.80	351.63	354.76	344.57				
Fuel Purchased lb.	515,250	339,900	484,800	258,500	152,200	49,250	52,000	68,200	66,600				
Cost of Fuel Purchased $	1,146.43	756.28	1,078.68	575.16	349.58	120.66	127.40	167.09	163.17				
Fuel Consumed lb.	745,250	429,900	455,800	228,500	147,600	77,850	69,000	51,200	86,600				
Cost of Fuel Consumed $	1,658.18	956.53	1,014.16	619.66	328.14	190.73	169.05	125.44	212.17				
Cost of Ash Removal $	36.00	36.00	30.00	18.00	12.00	6.00	12.00	None	12.00				
Total Cost of Supplies $	179.27	230.39	85.21	182.45	349.66	462.96	79.36	149.20	140.80				
Cost of Supplies to Power Plant $	81.52	106.18	15.81	56.12	179.39	264.76	55.86	49.20	34.45				
Cost of Supplies to Building Equipment Maint. $	97.75	124.21	69.40	126.33	170.27	198.20	23.50	100.00	106.35				
Fixed Charges $													
Income from Steam Sold $	1,283.96	924.66	959.08	669.83	462.41	312.76	341.92	217.41	108.79				
Other Income from Tenants $	36.24	52.00	84.55	210.24	84.86	77.38	74.93	32.00	78.74				
Gross Operating Expense $													
Net Operating Expense $													
Water Evaporated From and At 212°F. lb.	5,187,700	3,129,907	3,517,400	2,250,462	1,237,975	522,120	398,724	381,441	518,100				
Water Evaporated per lb. Fuel Consumed From and At 212°F lb.	6.961	7.28	7.74	8.08	8.387	6.7	5.778	7.45	6.00				
Cost Per M.lb. Steam Based on Cost of Fuel Plus Ash Removal cts.	32.65	31.7	29.6	28.3	28.33	37.68	45.4	32.9	43.2				
Cost Per M.lb. Steam Based on Gross Operating Expense cts.													
Cost Per M.lb. Steam Based on Net Operating Expense cts.													
Steam Sold lb.	1,105,000	708,300	801,900	552,000	362,100	231,300	253,000	236,900	111,700				
Steam Used to Heating Hot Water lb.	200,000	200,000	200,000	200,000	200,000	147,820	2,724	1,541	200,000				
Steam Used in Restaurant 81 E. Lake St. lb.	143,000	132,000	143,000	143,000	143,000	143,000	143,000	143,000	143,000				
Steam Used in Heating Lake-Michigan Bldg lb.	3,739,700	2,089,607	2,382,500	1,355,462	532,875				53,400				
Heating Surface in Lake-Michigan Bldg sq. ft.	30,820	30,820	30,820	30,820	30,820				30,820				
Steam Used per Square Foot Heating Surface lb.	121.3	68	77.3	43.9	17.3				1.7				
Cost of Heating per Sq. Ft. Heating Surface (Fuel and Ash) cts.	3.96	2.16	2.29	1.25	0.5								
Mean Out-Door Temperature °F.	20.84	37.9	37.23	48.66	62.16	70.86	74.9	73.8	70.7				
Cost Per M.lb. Steam Based on Cost of Fuel—Ash—Labor cts.	42.26	46.3	42.07	44.77	58.76	112.00	137.3	129.4	112.4				
Cost Per M.lb. Steam Based on Total Cost of Oper. (Fixed ch. ex.) cts.	43.79	50.73	42.52	47.27	72.39	162.7	151.2	142.6	119.1				
Total Travel all Elevators Car Miles	4,353.0	3,994.1	4,302.9	4,403.3	4,290.7	4,265.0	4,358.2	4,245.0	4,148.7				
K. W. Consumed K.W.	18,600	16,900	18,000	19,000	19,100	18,400	18,500	17,300	17,300				
Car Miles per K.W. Consumed Miles	.234	.236	.239	.2317	.2245	.2318	.235	.245	.239				
K.W. Consumed per Car Mile K.W.	4.04	4.231	4.183	4.312	4.451	4.314	4.015	4.075	4.17				
Water Used in Building C. F.	162,070	186,300	183,980	220,100	205,480	184,470	194,780	192,850	203,870				

Instruction Charts

"Instruction charts for men have been made up to show the advantage of maintaining as high CO_2 readings as possible. The men are enjoined to keep evaporation at high points. Another chart shows cost of steam per 1,000 pounds at various evaporations."

We made some special inquiry among power engineers who serve Chicago down town buildings and found that facts like these are seldom brought out in the average building that rents for profits. In Lake-Michigan building the cost per 1,000 pounds of steam in January (plus ash removal) was $.3265; September showed a cost of $.432. The cost of heating per square foot of heating surface (fuel and ash) was $.0396 in January and $.017 in September. Utilization is a factor. The cost per 1,000 pounds of steam including fuel-ash-labor was $.4226 in January and $1.124 in September.

Another observation of much interest in this building showed more than double costs for heating certain rooms where weather strips had been displaced or ineffectively installed—*S. P. M.*

Heating-Piping
and Air Conditioning

1895

American Society *of* Heating *and* Ventilating Engineers
JOURNAL SECTION

NOVEMBER *1930*

Material in this Section prepared by Publication Committee: G. L. Larson, *Chairman;* H. H. Angus, and W. A. Rowe : A. V. Hutchinson, Manager of Publications

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Utilization of Waste Heat

By **Perry West**[1], **Newark, N. J.**

MEMBER

WASTE heat should be used for heating and ventilating requirements wherever it is available. The sources of such waste heat include electric or power generation, industrial and condensing processes, and furnace gases.

Where heat and power or electric current are required, steam may first be used for generating the power or electric current and the exhaust from the generating units then used for heating. Likewise in many industrial processes requiring heat at certain temperatures, steam may first be passed through engines or turbines, for generating any power or electric current required and used for the processes at the pressure required to give the desired temperature.

The waste heat from blast furnaces, roasting ovens, annealing ovens, smelting cupolas, Bessemer converters, heat-treatment furnaces and other processes of the iron and steel industry may be used for operating boilers, heating air or water or for other purposes requiring heat. Gases and vapors from digesters in pulp mills, from dryers of all kinds, from kettles, vats, etc., may be used for heating in one form or another.

In connection with ventilation and air conditioning and especially where the gases and dusts from industrial processes are to be removed, collected or otherwise handled, it frequently occurs that there is considerable recovery value in the by-product material as well as in the heat wasted.

General Economic Aspects

Wherever waste heat is available for heating purposes, the question arises as to whether it can be used practically and economically. Each case is a separate problem which must be solved from an analysis of the conditions involved.

In general, there is an added first cost for equipment with its resultant fixed operating costs, added costs for labor and added costs (or at least energies) for management, wherever a department or operating unit is required for making use of waste products. The governing factors usually are the return on the investment and energies expended. There is no established rule for the monetary return required, since local conditions, such as freight rates, competition and relative returns on efforts in other departments of a business may make it advantageous for one company to save a small percentage in order to secure its share of business while another company might find it advantageous to expend its money and energies along different lines. However, it is usually considered that unless a system saves ten per cent over and above its carrying charges it is not an attractive proposition.

Heating with Exhaust Steam

Perhaps the most familiar example of the utilization of waste heat is the use of high pressure steam for generating power or electric current and the reuse of the exhaust steam for heating. The exhaust steam may be circulated directly through the radiators, heaters, air heaters, humidifiers or other elements of a heating, or of a heating and ventilating system, or it may be used for heating water which in turn may be circulated through similar elements of a hot water heating system. It may also be used for domestic water heating.

Frequently there is a fluctuating load on the power or electric generating apparatus which in turn produces fluctuating quantities of exhaust steam. The quantity of steam required for heating is also variable. These two variables do not, as a rule, coincide so that the total quantity of exhaust steam produced may be either equal to, a certain percentage less or a certain percentage more than the total quantity of steam required for heating, or for heating and ventilating either during the heating season or throughout the year. These relations change throughout the day and night and throughout the seasons of the year.

There are also wide variations in the heating requirements due to changes in weather conditions and wide variations in the exhaust produced due to such changes as in the demands for lighting on dark or light days, for elevator power during rush or slack periods, for air conditioning on good or bad days and for many other variables in the demands for power or light, depending on the character of building and its operating requirements.

Adjustment Between Heating Loads and Exhaust Steam

The way in which the quantities of exhaust steam produced and the quantities of steam required during the various intervals of time throughout the year coincide will determine how much may be used, how much must be thrown away and how much live steam must be used to supplement the exhaust steam deficit, unless some means is adopted for adjusting the supply to the demand.

One method of making this adjustment is to store exhaust steam or its equivalent in heat, when it is in excess of the demand, and feed this to the system when the exhaust being produced is below the demand. Hot water accumulators may be used for this purpose by storing heat in water to be fed to the boilers, in water to be used for domestic hot water service, in water to be used in a hot water heating system, in water to be used for process work, or for some combination of these purposes.

Another method is to adjust the quantity of exhaust produced somewhat in concordance with the heating demand. A bleeder turbine from which the exhaust may be taken for the lower heating demands and additional higher pressure steam from its successive stages as this demand goes up is frequently a satisfactory so-

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[1] Consulting engineer.

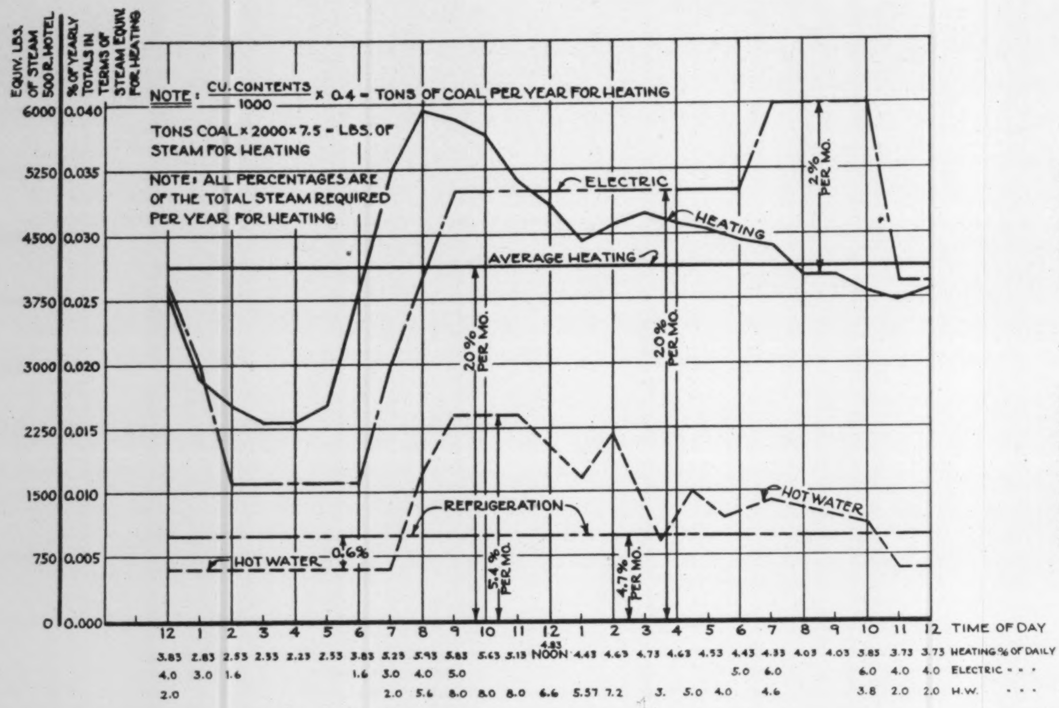

Fig. 1—January and February Hotel Load Curves in Terms of Steam Equivalents for Heating[a]

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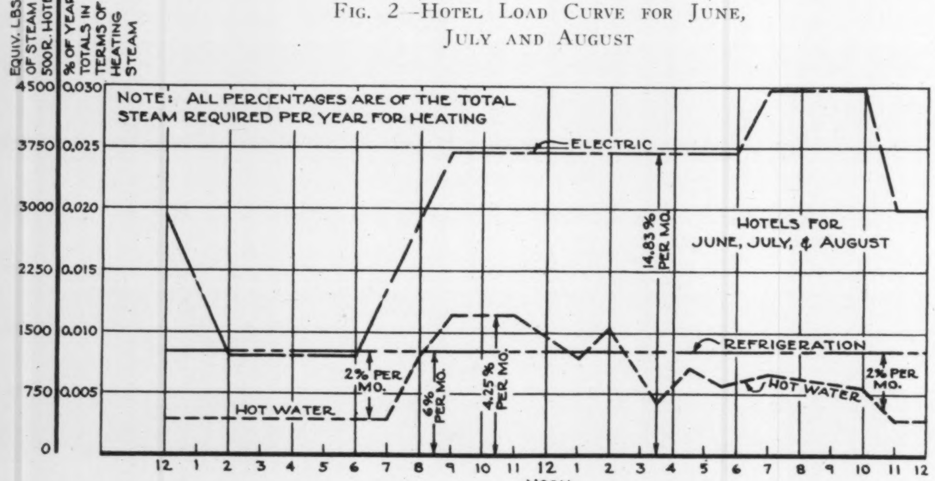
[a] Ventilation Steam Requirements (15%) not included. Generators at 25 lb per indicated horsepower per hour; 30 tons of refrigeration for 500 rooms at 25 lb per indicated horsepower per hour. No allowance for condensation (about 20 per cent).

lution for a case where the normal exhaust is deficient. If such a turbine is also arranged to run condensing under various controllable degrees of vacuum and arrangements are made for either circulating the exhaust through the heating system under various pressures above and below atmospheric pressure or for similarly using the exhaust for heating water, such an arrangement may adjust the exhaust to a wide variation of heating demands.

With such an arrangement, the exhaust supply may be automatically adjusted to the demand by having automatic control of the vacuum in the turbine from an outdoor thermostat operating on the flow of cooling water on the condenser and remote control valves in the bleeder steam supplies controlled from the same or a coordinate thermostat. This arrangement has certain advantages over that in which exhaust steam is discharged to the atmosphere when it is in excess of the demand and where live steam at boiler pressure is used to make up the deficit when the exhaust is insufficient. For example, the condensing of the excess exhaust steam increases the efficiency of the turbine so that the total heat rejected to the condenser is less than would otherwise be thrown away, while the use of the bleeder steam conserves heat by allowing a certain amount of power to be generated from this steam before it is fed to the heating system.

A modification of this method of adjustment, especially as between the heating and non-heating seasons, is to use engines or turbines which may be run condensing throughout the non-heating season and non-condensing throughout the heating season. Another modification is to have non-condensing engines or turbines for use during the heating season and low pressure exhaust condensing turbines for using the exhaust steam from these during the non-heating season. This latter arrangement affords a high efficiency since each unit operates under conditions for which it may be best designed. The low pressure turbine in this case may even be made a bleeder turbine and be arranged to operate on various degrees of vacuum with automatic control as previously stated. While this arrangement of units generally shows a high overall economy there is some disadvantage in installing condensing units which have to

Fig. 2—Hotel Load Curve for June, July and August



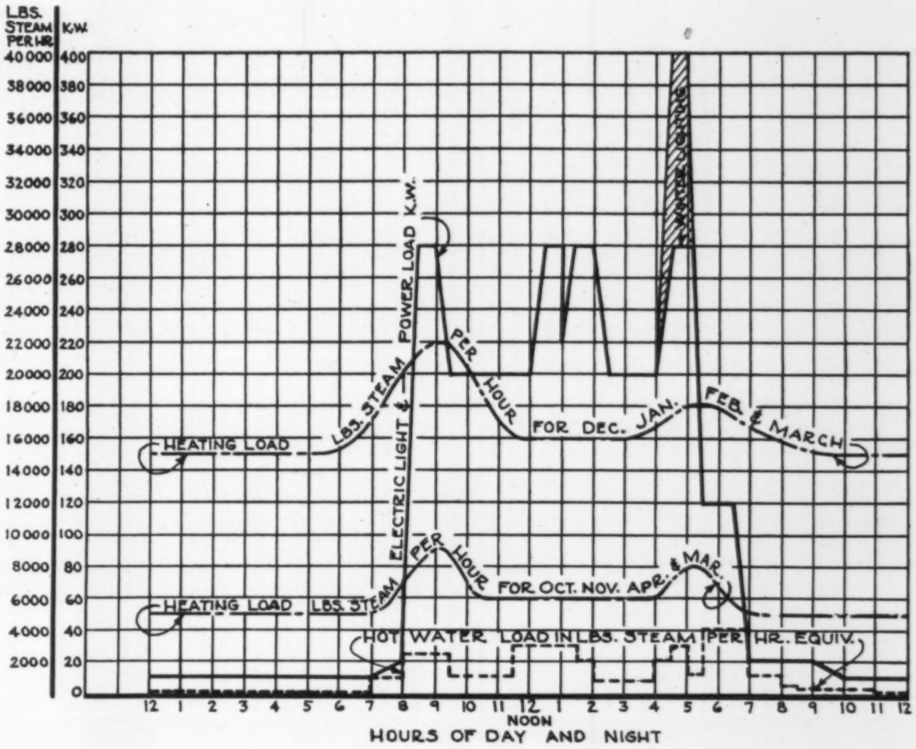


FIG. 3—OFFICE BUILDING LOAD CURVES

be frequently started up or shut down and by-passed and hence a straight bleeder turbine is frequently used as a matter of simplicity.

Still another method is to adjust the heating demand so that it will balance as nearly as possible the exhaust steam produced. The use of hand operated or automatic remote-control sectional, or zone, regulating valves for separately controlling the various sides and vertical sections of a building in accordance with the exposures, wind velocities, sunshine, etc., the use of automatic temperature control, fractional distribution of steam, distribution of steam under varying degrees of vacuum and the use of forced hot water with temperature control, all tend to keep the demand commensurate with the actual weather requirements, thereby reducing the maximum demand and at the same time the amount of fluctuation by supplying heat only where it is needed.

Heating Load and Amount of Exhaust Steam Available

In all of these adjustments the question of relative economies and the cost to produce them must necessarily enter into the problem. The efficiencies of engines and turbines vary with the types, sizes, speeds, load conditions and with the steam pressure, temperature, back pressure or vacuum under which they operate. Actual guaranteed water rates for the units and conditions to be employed on any particular operation should be obtained from several reputable makers before endeavoring to estimate the exhaust.

Load conditions and heating requirements must then be computed for the various hours of the day and night and for the various days of the year. Generally, a 24-hour day load and heating requirement chart for each of the four seasons of the year will be sufficient although in many cases the load may be different for some days of each week and of course over holidays and other non-operating periods.

Typical load curves are shown by Figs. 1, 2, 3, and 4. In the hotel diagrams (Figs. 1 and 2) it will be noted that the steam required for generating the refrigeration power is shown separately and is not included in the steam required to generate the remaining electric current or power. This is for the purpose of comparing it with the hot water load, which it practically equals in all of these charts. For this reason the refrigerating apparatus in this class of building may frequently be steam driven and the exhaust thus produced be used with good economy for heating the domestic hot water even where the remainder of the electric current is purchased from the outside and steam or hot water direct from the boiler plant is used for heating.

Procedure for Making Analysis

In making an analysis of any particular operation it is necessary to develop charts similar to Figs. 1, 2, 3

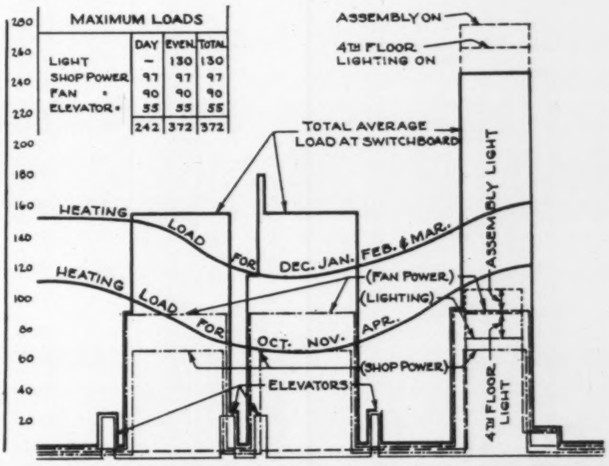


FIG. 4—HIGH SCHOOL LOAD CURVES

TABLE 1—APPROXIMATE FULL LOAD EFFICIENCIES FOR ENGINE AND MULTI-STAGE IMPULSE STEAM TURBINE-GENERATORS—GEARED TYPE

RATED CAPACITY AT FULL LOAD KW.	MECHANICAL EFFICIENCIES OF ENGINE UNITS	MECHANICAL EFFICIENCIES OF TURBINE UNITS	BRAKE POTENTIAL EFFICIENCY RATIO OF TURBINES	(a)FACTORS FOR CONVERTING IDEAL WATER RATES TO ACTUAL WATER RATES (FOR TURBINES ONLY)	GENERATOR EFFICIENCIES AT FULL LOAD
100	0.900	0.930	0.45	2.22	0.932
200	0.905	0.935	0.50	2.00	0.934
300	0.910	0.940	0.54	1.85	0.936
400	0.915	0.942	0.58	1.74	0.938
500	0.920	0.945	0.61	1.64	0.940
600	0.925	0.947	0.64	1.56	0.942
750	0.930	0.950	0.65	1.54	0.944
1000	0.935	0.952	0.67	1.49	0.945
1250	0.940	0.955	0.67	1.49	0.947
1500	0.945	0.957	0.67	1.49	0.950
2000	0.950	0.960	0.68	1.45	0.954
CONDENSING TURBINES					
500		0.950	0.68	1.45	0.940
600		0.952	0.69	1.44	0.942
750		0.954	0.70	1.43	0.944
1000		0.956	0.70	1.43	0.945
1250		0.958	0.70	1.43	0.947
1500		0.960	0.70	1.43	0.950
2000		0.960	0.70	1.43	0.954
				(For Engines only)	
Single valve automatic engine units up to 400 kw				1.64	
Four valve automatic engine units up to 400 kw				1.42	
Unaflow engine units up to 400 kw.............				1.20	

[a] These factors are for indicated engine or brake turbine loads and do not take into account the mechanical efficiency of engine units or generator efficiencies for either engine or turbine units; but do include the mechanical efficiency of turbines.

and 4. The total steam required for heating per square foot of direct radiation, or its equivalent, per season, for different classes of buildings is as follows:[2]

For manufacturing or commercial loft type, where steam is used to heat the premises during the day hours to maintain 65 to 68 F from 9 A. M. to 5 P. M. No Sunday or holiday use and no night use. Factor: 325 lb. per square foot of radiation per season.

For office buildings using steam during daylight hours only to maintain 70 F from 9 A. M. to 6 P. M. for approximately 240 days (heating season). No night use. Factor: 400 lb. per square foot of radiation per season.

For office buildings using steam during day hours and at night when required to 7, 8 and 9 P. M. (customary where there are stock brokers or banking offices), 204 days. Factor: 500 lb. per square foot of radiation per season.

For residences of the block type (not detached) where high-class heating service is required somewhat similar to apartment buildings. *Factor*: 550 lb. per square foot of radiation per season.

For apartment houses where high-class heating service is required. (Steam off at midnight.) Factor: 650 lb. per square foot of radiation per season.

For hotels (commercial type) where very high-class service is required; 24-hour service. Factor: 800 lb. per square foot of radiation per season.

Of the total steam required for any particular service, the following percentages of the whole may be apportioned to the various months of the heating season in the colder climates: January, 20 per cent; February, 20 per cent; March, 15 per cent; April, 10 per cent; May, 5 per cent; October, 6 per cent; November, 8 per cent; December, 16 per cent.

To determine the amount of exhaust steam available, it is necessary first to develop electric current or power load curves and from these to estimate the amounts of steam used during various periods. Fig. 5 gives the ideal water rate for engines or turbines working under different initial pressure and back pressure or vacuum conditions, with corrections for superheat in the initial steam. Table 1 gives the factors by which ideal water rates may be multiplied to obtain average actual expected water rates for engines and turbines of various capacities when operating under full load conditions. Table 2 gives the factors by which the actual water rates under full load operating conditions should be multiplied to obtain actual expected water rates under various load conditions. From the load curves and these tables it is a simple matter to develop curves showing the steam required by the electric generating or power units.

The next step is to determine the amount of heat available in this steam for heating, etc., after it has passed through the generating units. This can be done by deducting the heat transferred into work, plus that lost by radiation from the heat in the steam entering the generating units. Since 2,546 Btu per hour = one horsepower hour and 3,415 Btu per hour = one kilowatt hour the heat extracted in performing work per pound of steam used may be obtained by dividing these figures by the water rates of the units per indicated horsepower hour or per kilowatt hour, respectively, of work done by the steam. From 1 per cent to 2 per cent may be

TABLE 2—FACTORS BY WHICH TO MULTIPLY ACTUAL WATER RATES AT FULL LOAD TO GET ACTUAL WATER RATES AT FRACTIONAL LOADS ON ENGINE AND TURBINE DRIVEN ELECTRIC GENERATING UNITS[a]

SIZE AND KIND OF UNIT	LOAD			
	¼	½	¾	FULL
Single valve engines up to 400 kw..	1.32	1.09	1.03	1.00
Four valve engines up to 400 kw...	1.30	1.08	1.03	1.00
Unaflow engines up to 400 kw.....	1.10	1.02	0.980	1.00
Geared turbines up to 600 kw......	1.57	1.25	1.05	1.00
Geared turbines 600 to 2000 kw....	1.50	1.20	1.04	1.00

[a] These values are for indicated or brake loads and do not take into account variations in generator efficiencies under various load conditions. Generator efficiencies vary from 85 per cent at one quarter load to 95 per cent at full load.

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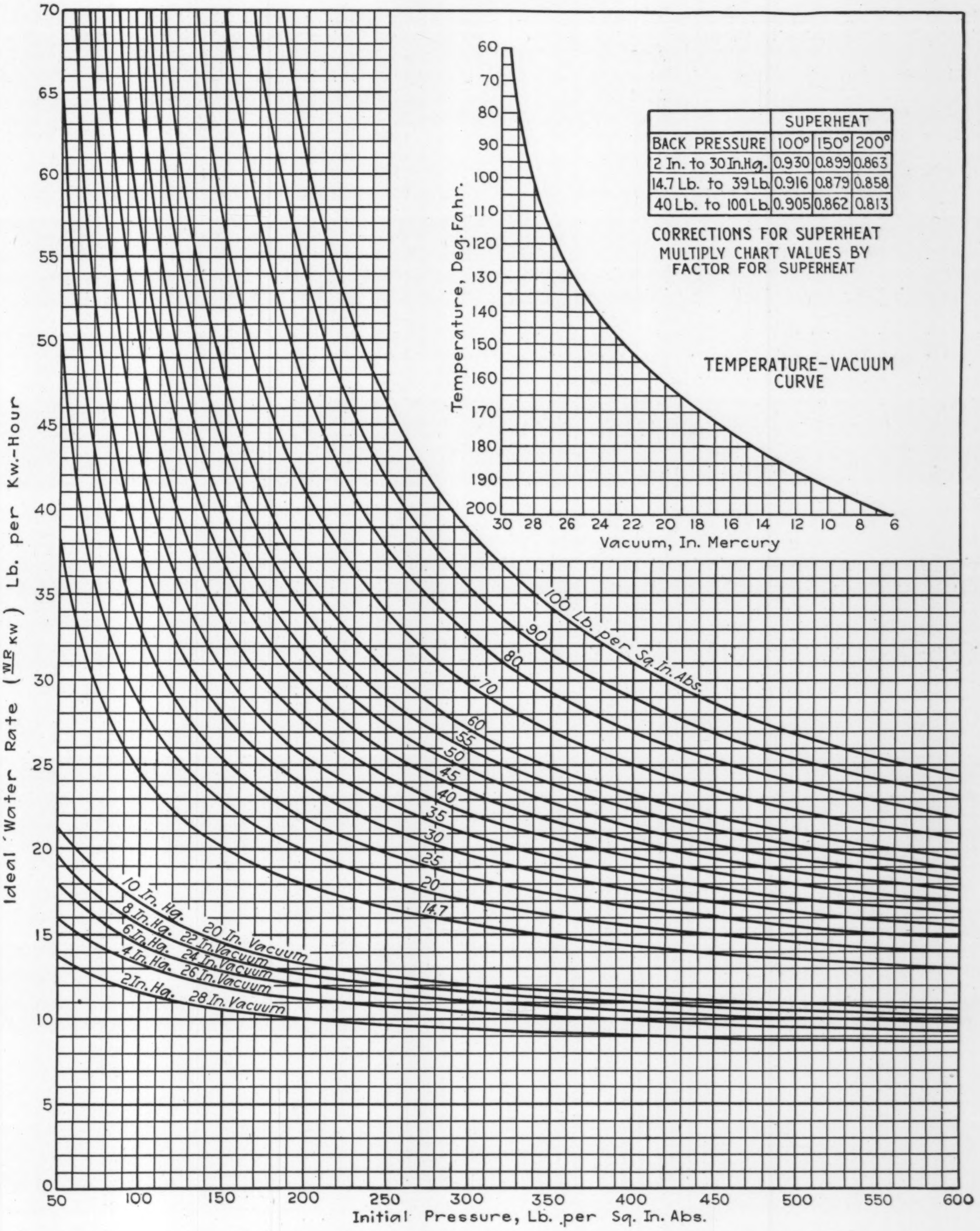
[2] See Chapter 24, A. S. H. V. E. GUIDE, 1931.



	SUPERHEAT		
BACK PRESSURE	100°	150°	200°
2 In. to 30 In. Hg.	0.930	0.899	0.863
14.7 Lb. to 39 Lb.	0.916	0.879	0.858
40 Lb. to 100 Lb.	0.905	0.862	0.813

CORRECTIONS FOR SUPERHEAT
MULTIPLY CHART VALUES BY
FACTOR FOR SUPERHEAT

FIG. 5—IDEAL WATER RATE PER KILOWATT HOUR WITH DRY SATURATED STEAM FOR VARIOUS INITIAL PRESSURES AND BACK PRESSURES[a]

deducted for radiation and other heat losses, depending upon the size of unit.

Example. A 200 kilowatt turbine is operating on saturated steam at 150 lb. gage pressure and consuming 40 lb. of steam per kilowatt hour delivered, at full load. Determine the heat available in the exhaust for heating.

Solution. The mechanical efficiency of the turbine is 0.935 and the efficiency of the generator is 0.934 (See Table 1), giving a combined efficiency of 0.935 × 0.934 = 0.873 which would mean 40 × 0.873 = 34.92 lb. per kilowatt hour of external work actually done by the steam. The total heat of the steam under the conditions as fed to the turbine is 1,195 Btu per pound.

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[a] For ideal water rate per indicated horsepower hour, divide these values by 1.34.

The heat extracted in external work $= \frac{3{,}415}{34.92} = 97.74$ Btu per pound of steam. To this add 1½ per cent for heat lost by radiation, or $97.74 + 1.47 = 99.21$ Btu which is the total heat loss per pound of steam in passing through the turbine. This leaves $1{,}195 - 99.21 = 1{,}095.79$ Btu in each pound of the exhaust.

If the exhaust is to be used and condensed at atmospheric pressure there will be a further deduction of 180 Btu per pound for the heat of the liquid at this pressure. This leaves $1{,}095.79 - 180 = 915.79$ Btu per pound of steam fed to the turbine as being available in the exhaust for heating at atmospheric pressure.

In a similar manner the heat available per pound in the exhaust from an engine driven unit for heating may be estimated by dividing the heat equivalent per unit of work by the number of pounds of steam fed to the unit per unit of actual external work done, adding to this from 1 per cent to 2 per cent for radiation, and subtracting this sum from the total heat in a pound of the steam as fed to the unit and then subtracting from this remainder the heat per pound in the condensate or rejected steam leaving the heating system.

In the case of an engine-driven unit, the heat equivalent of a horsepower hour (2,546 Btu) should be divided by the water rate per indicated horsepower. If the water rate is given per kilowatt hour delivered by the generator, this may be reduced to pounds per indicated horsepower hour by dividing by 1.34 and multiplying by the mechanical efficiency of the unit and the efficiency of the generator.

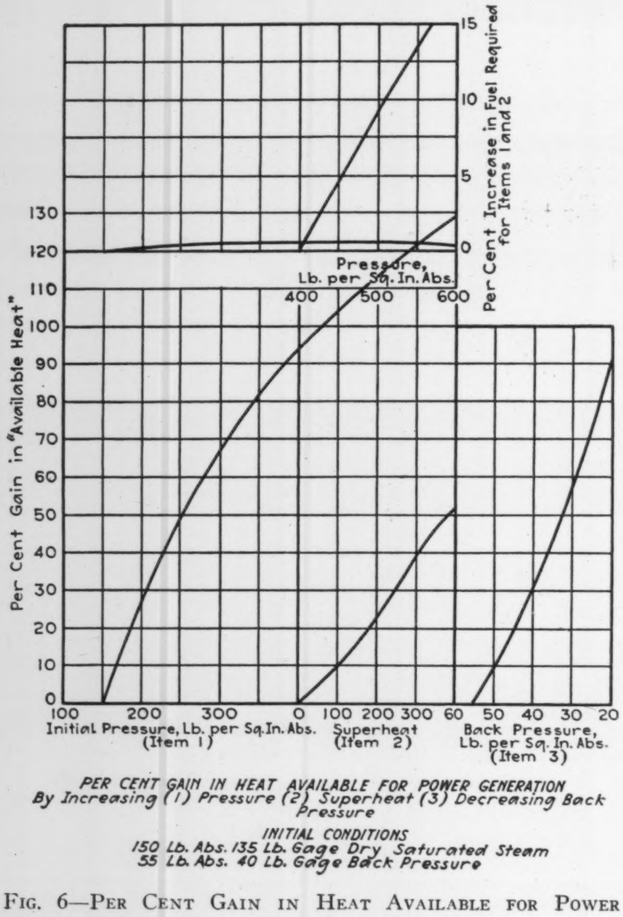


FIG. 6—PER CENT GAIN IN HEAT AVAILABLE FOR POWER GENERATION

Example. A 200-kw., engine-driven unit is operating under the same conditions as the turbine unit in the preceding example, and delivering one kilowatt hour on 37.8 lb. of water.

The mechanical efficiency of the unit is 0.905 and the efficiency of the generator is 0.934 (See Table 1). The water rate per indicated horsepower is therefore

$$\frac{37.8}{1.34} \times 0.905 \times 0.934 = \frac{37.8}{1.34} \times 0.845 = 23.84$$

The heat equivalent of the external work done per pound of steam is $\frac{2{,}546}{23.84} = 106.79$ Btu. To this add 1½ per cent for radiation, or $106.79 + 1.60 = 108.39$ Btu which is the total heat lost per pound of steam in passing through the engine. Deduct this from the total heat per pound in the steam fed to the engine, or $1{,}195 - 108.39 = 1{,}086.61$ Btu.

Power from Process Steam

Wherever steam is employed for process work the boiler pressure is determined by that part of the process requiring the highest temperature, and generally the bulk of the steam is employed at a considerably lower temperature and corresponding pressure. The heat available per pound of available process steam, for the generation of power is the difference between the heat content of the steam at the generating pressure (initial condition) and the heat content of the steam corresponding to the pressure at which it is employed in the process (final condition), assuming adiabatic expansion (Rankine Cycle). This difference multiplied by the pounds of steam so employed per hour gives the total heat available for power generation by means of reciprocating engines or steam turbines.

The more or less fixed amount of process steam available at the existing process pressure is often found to be inadequate to supply all of the power requirements and some means should be adopted to secure as near a balance between the heat available and the heat equivalent of the power requirements as it is practical to obtain with a minimum amount of expense for extra fuel. The heat available per pound of steam may be increased by (1) increasing the initial pressure; (2) superheating, and (3) decreasing the process pressure (back pressure).

Fig. 6 shows an example of the comparative effect on the basis of per cent gain in the heat available per pound of steam of the foregoing items with an existing condition of 150 lb. per square inch absolute pressure (135 lb. gage) boiler pressure as required for a small part of the process dry saturated steam and 55 lb. per square inch absolute (40 lb. gage) pressure for the bulk of the process steam assumed available for the generation of power. The per cent gain is the difference between the available heat per pound of steam for the existing condition and the proposed condition divided by the available heat per pound for the existing operating condition.

Suppose, for example, there was found, on comparing the available process steam load and the power load curves and assuming a certain prime mover to be employed, a deficiency of approximately 15 per cent of the steam available for power generation. Fig. 6 indicates that this deficiency can be met in any one of following three ways:

1. Increasing the boiler pressure to 175 lb. per square inch absolute or an increase of 25 lb. per square inch.

2. Superheating the steam 150 deg.
3. Reducing the process (back) pressure to 47 lb. per square inch absolute or a reduction in the process pressure of only 8 lb. per square inch.

Frequently higher process pressures than necessary are carried and some experimenting to this end may return good dividends. The natural procedure in the investigation is to start in the reverse order, and it is frequently found that at least two or possibly all of these gains must be employed to secure the desired result. There remains the possibility of increasing the amount of power that may be generated without additional expenditure for fuel by substituting electric motors for the small steam engines or steam pumps employed in the plant.

The possibility of effecting boiler plant economies, tending to offset any additional fuel as may be required when items (1) and (2) are considered, should not be overlooked. A study of the existing plant requirements covering both the steam and electric power load curves may result in suggested methods to reduce the peak loads resulting in savings that likewise cost nothing to secure except the time required for the survey and analysis.

A number of assumptions are necessary to determine the most economic initial pressure and superheat for the required back pressure for the particular process involved. Some of the many items frequently considered in this connection are, the installation of superheaters for the existing boilers, of additional boiler capacity to be operated at a higher pressure in conjunction with straight non-condensing turbines, mixed pressure, extraction or bleeder type turbines, heat exchangers, steam regenerators or accumulators, and condensing operation utilizing hot well water as a source of hot water supply.

Amount and Conditions of Exhaust Steam for Process Use

The foregoing data on water rates and heat extraction of engines and turbines may be used for determining the amount and condition of the exhaust steam from power units for process use. The condition of the steam at exhaust from a turbine may be determined with sufficient accuracy for all practical purposes as follows: Determine the total heat drop in the steam, between initial and exhaust condition, from a table of properties of steam (including superheat if any) and after multiplying this by the brake potential efficiency ratio of the turbine (Table 1), subtract the result from the total heat in the initial steam and the remainder will be the total heat left in the exhaust; then compare this with the total heat of saturated steam at the exhaust pressure or with a Mollier diagram for determining the quality of the steam at exhaust.

Referring to Fig. 7, it is assumed that the steam is supplied to the turbine throttle at a pressure of 175 lb. per square inch absolute, as indicated on the diagram by *a*. The loss in pressure through the throttle valve is assumed at 15 lb. per square inch, so that the steam enters the first pressure stage of the turbine with the same heat content at 160 lb. per square inch absolute indicated at *a′* with a temperature of 518 F (156 deg superheat). If the steam expanded adiabatically from 160 lb. per square inch to an absolute pressure of 35 lb. per square inch at point *b′*, the available heat (*h*) would be the difference between the total heat for conditions *a′* and *b′* or $h = 1{,}282\ (i_1) - 1{,}149\ (i_2) = 133$ Btu per pound of steam.

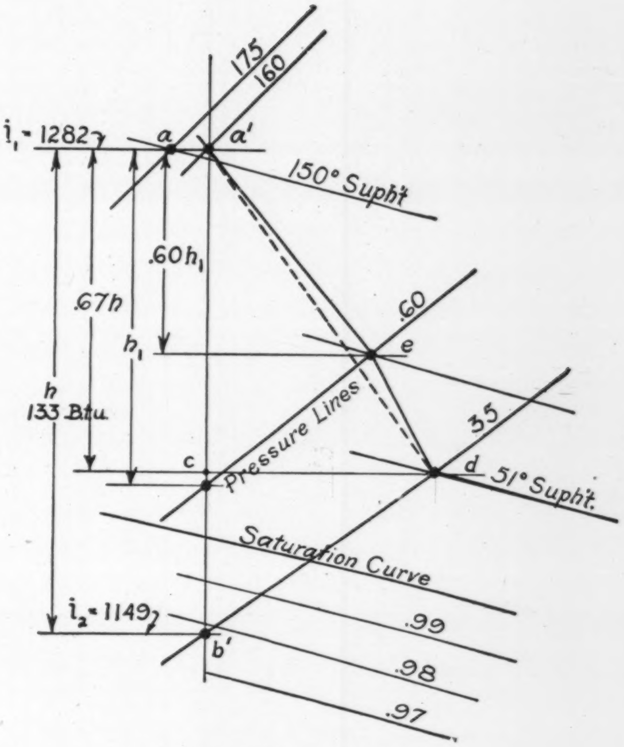


Fig. 7—Mollier Diagram

a. State of Steam Entering Throttle Valve, 175 lb Per Square Inch 150 Deg Superheat.
a′. State of Steam for 15 lb Drop in Pressure Through Valve, 160 lb Per Square Inch Absolute 156 Deg Superheat.
b′. Theoretical State After Expanding from 160 lb. Absolute to 35 lb Absolute Pressure.
d. Final Estimated State of Exhaust from Turbine, 35 lb Absolute, 318 F or 59 Deg Superheat.

For an assumed turbine brake potential efficiency of 0.63 and a generator efficiency of 0.944 at full load, the expected water rate per kilowatt hour =

$$E\,W\,R_{\mathrm{kw}} = \frac{2{,}546 \times 1.34}{0.63 \times 0.944 \times 133} = 43 \text{ lb. per kilowatt hour.}$$

Fig. 7 shows the estimated expansion line *a′-e-d* for a two-pressure three-velocity stage machine. It is assumed that the first pressure stage is a Curtis type stage and that ϕ_t for the first stage is 0.64. For an assumed mechanical efficiency of 0.94, the efficiency for the complete turbine is

$$\phi_1 = \frac{\phi_t}{0.94} = 0.67$$

Use of the Willans Line

The well known principle of the Willans line may be made use of for determining the total quantity of steam required for the varying loads on an engine or turbine as follows:

Referring to the diagram located in the upper left hand corner of Fig. 8, if the total pounds of steam for any two loads, as for example, no load (*b*) and full load (*a*) are known for a throttling steam engine or a steam

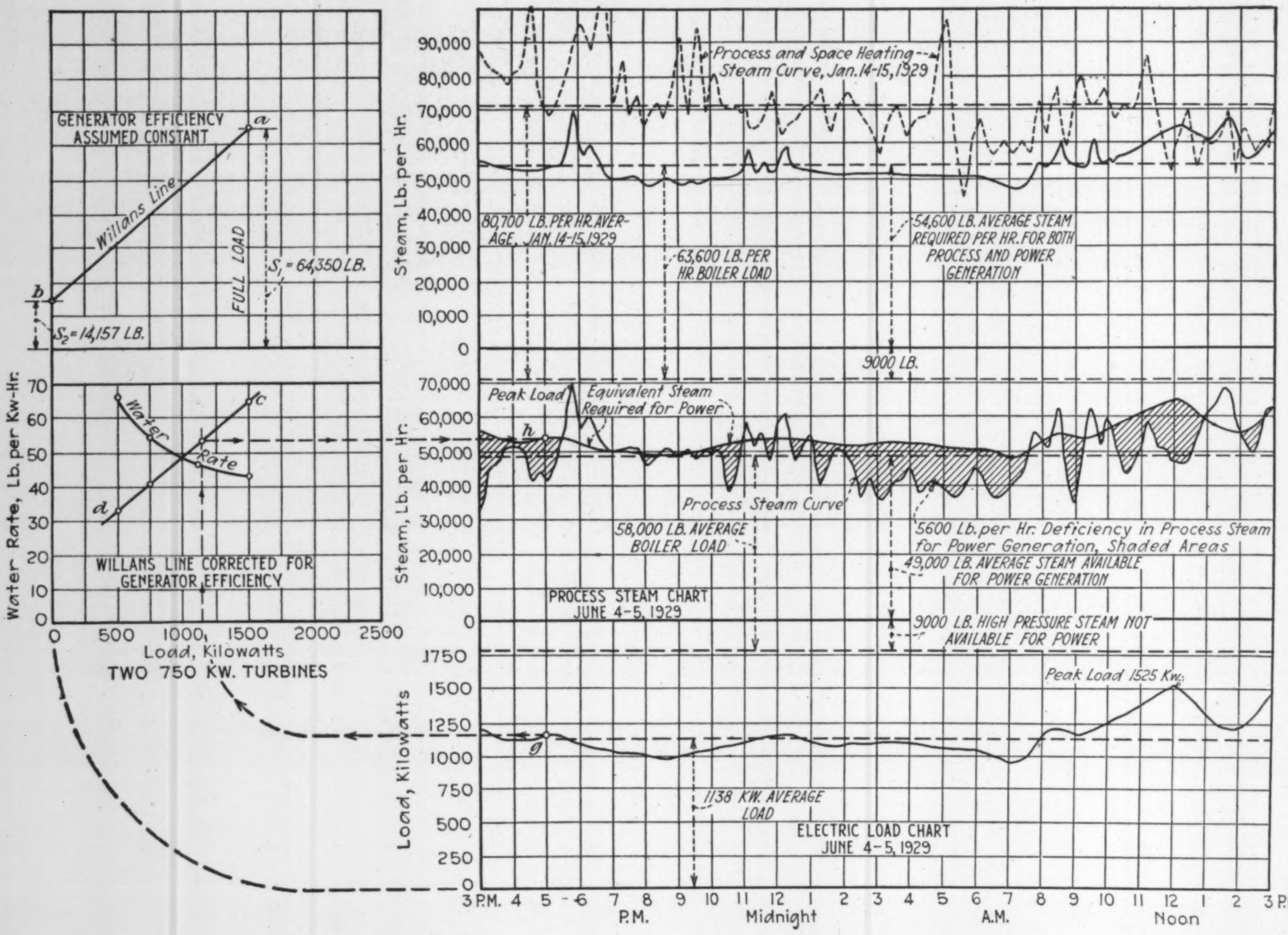


FIG. 8—STEAM AND ELECTRIC LOAD CURVES FOR AN INDUSTRIAL PLANT

turbine, and a straight line is drawn between these points, the total pounds of steam for any intermediate load is correctly given by the ordinate for that load. The diagram is correct when the rating is based on the brake horsepower of a turbine or the indicated horsepower for an engine. The total steam for any intermediate load may also be calculated by means of the following formula:

where $S = S_1 F (1 - y) + y.$ (1)

S = Total steam per hour for the desired load.
S_1 = Total steam per hour for the full load.
y = Ratio of no load steam to full load brake horsepower steam.
= Approximately 0.22 for non-condensing turbines.
= Approximately 0.14 for condensing turbines.
= Approximately 0.16 for low pressure condensing turbines (atmospheric pressure —28 in. vacuum).
F = fraction of full load corresponding to S.

It is necessary to correct the Willans line to allow for a decreasing generator efficiency as the load is reduced below the power developed at full load. The water rate for any load is determined by dividing the total steam per hour by the corresponding load.

Example of Power from Process Steam

Fig. 8 illustrates a method employed in plotting an equivalent electric power load curve from an available process steam load curve, or an equivalent steam load curve from a power load curve. It is, of course, desirable to have the actual guaranteed water rates from the several manufacturers, whose machines are being considered, covering the proposed conditions of operation. These water rates will ordinarily be found to vary somewhat between the different types of machines proposed. The correct use of the data herein presented, however, will, it is believed, give sufficiently accurate water rates for the purpose intended. The process steam load curve and the purchased electric power load curve shown were plotted from meter readings on an industrial plant.

Steam Load Curve

The steam load curve of this plant is shown by the dashed line at the top of Fig. 8. The boilers are designed for a pressure of 175 lb. per square inch gage and are not equipped with superheaters. High pressure steam for certain operations is required at the average rate of approximately 9,000 lb. per hour, and is fairly constant. The remainder of the steam generated, or 49,000 lb. average per hour, it is assumed, will satisfactorily supply the remainder of the requirements (cookers, evaporators, drying, water heating, etc.) with a back pressure at the turbines of 20 lb. per square inch gage. In order to supply dry steam at the exhaust outlet of the turbines it is proposed to install superheaters for all of the existing boilers. The cost of 13,000 Btu coal in the bunkers is $3.75 per ton. The average boiler and grate efficiency of this plant is 68 per cent. From an inspection of the electric power load curve it is evident that the full load turbine rating required should be approximately 1,500 kw. to provide for the peak load. Probably the most practical combination, all things considered, would be the installation of three 750 kw. machines, one of which would be a spare unit.

Assuming an initial throttle pressure at the turbine of 160 lb. per square inch gage (175 lb. absolute), 150 deg. superheat and 15 lb. drop through throttle with 20 lb. per square inch gage (35 lb. absolute) back pressure, the ideal water rate with 160 lb. absolute pressure is:

$W\ R_{kw} = 29.2 \times 0.879$ (superheat correction) $= 25.66$ lb. per kilowatt hour.

Assuming a potential efficiency ratio for the turbine of 0.634 and a generator efficiency of 0.944, the expected full load water rate for a 750 kw. turbine-generator set for the assumed condition is:

$$E\ W\ R_{kw} = \frac{25.66}{0.634 \times 0.944} = 42.9 \text{ lb per kilowatt hour.}$$

A guaranteed water rate approximately 10 per cent lower than this figure can be obtained from some manufacturers.

The total estimated steam required at full load for two 750-kw. units is

$$2 \times 750 \times 42.9 = 64{,}350 \text{ lb. per hour.}$$

It is assumed that the ratio of no load to full load steam is $y = 0.22$, the estimated steam at no load being equal to $0.22 \times 64{,}350$ or 14,157 lb. per hour. The Willans line *a-b* may now be constructed as previously indicated and is evidently based on the assumption that the generator efficiency remains constant for all loads. The corrected total steam per hour, line *c-d,* is shown with the diagram directly below the Willans line and was plotted from the calculated data given in Table 3 which in turn is based on equation (1). The corrected total steam per hour for the various loads shown was obtained by multiplying the values as determined from the Willans line by the ratio of the generator efficiency at full load to the generator efficiency at the fractional load.

Any point, as *g* on the electric power load, may be translated into equivalent steam load by simply projecting over to the intersection with the total *steam per hour* line and transferring the equivalent power load to the steam chart, as indicated at *h*. Any point on the *available steam chart* curve may be translated into equivalent electric power by simply reversing this process. The deficiency in the process steam at various times during the day for generating the electric power required is shown by the shaded areas on the process steam chart (Fig. 8). Integrating these areas, the average deficiency in process steam to generate the power required is found to be 5,600 lb. per hour.

TABLE 3—VALUES OF STEAM PER HOUR, CORRECTED TOTAL STEAM PER HOUR AND WATER RATE CALCULATED FROM FORMULA (1)

LOAD KW.	FRACTIONAL LOAD	GENERATOR EFFICIENCY	RATIO OF GENERATOR EFFICIENCY FULL LOAD —— GENERATOR EFFICIENCY FRACTIONAL LOAD	STEAM PER PER HOUR FROM WILLANS LINE —— LB.	CORRECTED TOTAL STEAM PER HOUR —— LB.	WATER RATE KW. HOUR —— LB. E W R kw.
500	0.33⅓	0.890	1.060	31,500	33,390	66.8
750	0.50	0.909	1.038	39,500	41,001	54.6
1125	0.75	0.931	1.014	51,800	52,525	46.4
1500	1.00	0.944	1.000	64,350(S_1)	64,350	42.9

The average process steam available is 49,000 lb. per hour as determined by integrating the area under the process steam load curve. The average steam generated by the boilers for all purposes is 58,000 lb. per hour and the average required, if all power were to be generated at the plant, would be $58{,}000 + 5{,}600 = 63{,}600$ lb. per hour. The hourly quantities of steam as would be required, if all the power were to be generated, is shown by the steam load curve chart at the top of the figure. This chart shows that the peak load on the boiler plant would remain at 78,000 lb. per hour, and is in no way affected by the problem of power generation. The factor of evaporation for 160 lb. gage dry saturated steam and 200 F feed water is 1.0592 and for the same pressure condition with 150 deg. initial superheat is 1.148. The extra fuel required per hour, if all the power is to be generated, is, therefore:

$$\frac{[(63{,}600 \times 1.148) - (58{,}000 \times 1.059)] \times 971.7}{0.68 \times 13{,}000} = 1{,}274 \text{ lb.}$$

or

$$\frac{1{,}274 \times 24}{2{,}000} = 15.28 \text{ tons per day.}$$

The exhaust from the turbine will have a superheat of approximately 59 deg. (Mollier Diagram Fig. 7). Dry steam will therefore be supplied by the turbines to the process.

The deficiency in process steam could be practically eliminated if the average water rate of the turbines employed could be reduced to $\frac{49{,}000}{1{,}138}$ or 43.1 per kilowatt hour. The estimated average rate for the assumed conditions is approximately 49 lb. per kilowatt hour, which would require a reduction in the full load water rate of approximately 6 lb. This would require an ideal water rate of 20.9 lb. which corresponds to an initial pressure of 250 lb. absolute or 235 lb. gage. This increase in pressure would require a new boiler installation, which is not possible in this plant.

The estimated cost of the additional equipment required, if all the power is to be generated at the plant, is as follows:

3 750-kw. steam turbine generating sets	$51,000
3 Foundations for above machines	1,500
Superheater	8,500
Piping and electric wiring	17,000
Turbine house	12,000
Total	$90,000

The daily cost of producing electric power based on the preceding data is estimated as follows:

Fixed charges: 15% of $\frac{\$90{,}000}{365}$	$37.01
Additional fuel required, 15.28 tons at $3.75	57.30
Additional labor (stationary engineers)	15.00
Additional supplies	4.00
	$113.31

The average electric power load for the day is 1,138 kw. The estimated cost per kilowatt hour to produce this amount of power is therefore:

$$\frac{113.31}{24 \times 1{,}138} = \$0.00415$$

The cost of purchased power at this plant averages \$0.007 per kilowatt hour. This company would apparently save at least 40 per cent of the present power bill, or approximately \$22,000.00 per year, if they generated their own power. This saving represents a return of nearly 25 per cent on the estimated investment involved.

The amount of extra fuel, as previously estimated, may be reduced by the employment of extraction or bleeder type turbines, by passing the extra steam as required to generate the deficiency in power due to the lack of sufficient process steam through more turbine pressures stages to a final lower back pressure. It is evident that the extra steam required must be wasted so that any reduction in the amount results in a direct saving in fuel.

It is considerably more tedious to plot the equivalent steam chart, when a bleeder turbine is under consideration, to determine the deficiency in process steam. This is due to the fact that an extraction line is required corresponding to each amount of steam it is desired to transfer into electric power or vice versa. The following method is only a rough approximation, but is sometimes employed for preliminary estimates:

Assuming that the extra steam required is expanded in the turbine down to a back pressure of 17 lb. absolute (2 lb. gage) the heat available (Mollier diagram) is 187 Btu per pound. The heat available for 35 lb. absolute back pressure, as previously determined, is 133 Btu per pound. Therefore $1 - \frac{133}{187}$ or 0.30 (approximately) of the excess steam, as previously calculated may be saved by the use of bleeder type turbines, or 0.30 × 5,600 = 1,680 lb. per hour (average) and an estimated saving in fuel of 2.5 tons of coal per day. The saving in fuel per year is 2.5 × 365 × \$3.75 or \$3,421.88. The extra cost of the bleeder turbine equipment is approximately \$10,000.00. The fixed charges on this extra cost are \$1,500.00, so that the estimated net accrued saving is a comparatively small amount (\$1,921.88).

For more accurate methods of determining the results from use of bleeder or extractor turbines, also for water heating with exhaust steam for process work and for space heating with exhaust steam versus hot water, see Power from Process and Space Heating Steam, by L. A. Harding. [A. S. H. V. E. JOURNAL SECTION (*Heating, Piping and Air Conditioning*), November, 1929.]

Chart for Estimating Infiltration Heat Loss

Because of the influx of cold air by infiltration, provision must be made for the heat required to warm this air to room temperature. The heat required is given by the following equation:

$$H_1 = 0.24\ Q\ d\ (t - t_o) \qquad (1)$$

where

H_1 = Btu per hour required for heating air leaking into building from outside temperature t_o to inside temperature, t.

Q = cubic feet of air entering per hour at inside temperature, t.

d = density (pounds per cubic feet) of air at inside temperature, t.

t = inside temperature at the proper level.

t_o = outside air temperature for which heating system is designed.

0.24 = specific heat of air.

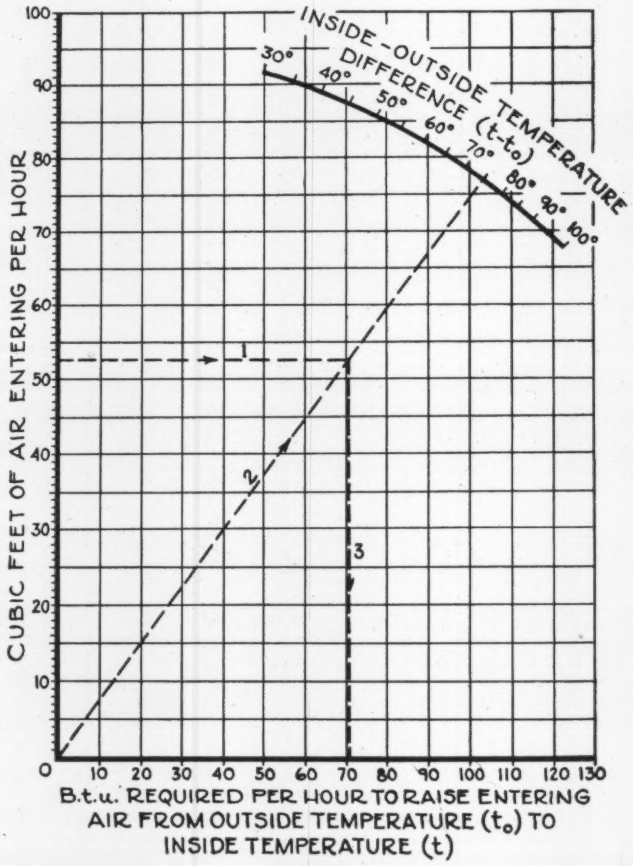
It is sufficiently accurate to take $d = 0.075$ lb. in which case the equation reduces to

$$H_1 = 0.018\ Q\ (t - t_o) \qquad (2)$$

The heat required for warming up the air entering by infiltration may be estimated by means of the accompanying chart. To use this chart, locate the air leakage through walls or through cracks around windows or doors (see Chapter 2, A. S. H. V. E. Guide 1930), on the left-hand scale, and draw a line horizontally to the right. Next find the temperature head on the upper scale and draw a line to the lower left-hand corner of the chart. At the intersection of the first two lines, draw a line vertically to the lower scale, on which is indicated the heat required in Btu per hour to raise the air entering by infiltration from the outside temperature t_o to the inside temperature t.—*P. D. C.*

New Hand Book Scheduled

A new edition of the Handbook of Domestic Oil Heating is scheduled to be published about January 1, 1931, according to the *American Oil Burner Association.* It is proposed to cover commercial and industrial installations as well as domestic installations in this new book.



Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces

By **L. W. Schad**[1], **East Pittsburgh, Pa.**
NON-MEMBER

AIR becomes ideal heat insulation if convection by it and radiation through it are prevented or greatly reduced. If, for instance, bright metal reflecting screens be placed in a heat path through air,[2] the transfer both by radiation and convection should be reduced. This paper contains information relative to the value of such screens in reducing thermal losses.

Two quite different test methods were employed for the purpose of obtaining data on the thermal transmission of walls built up of air spaces separated by bright metallic screens. In addition, some work was done to determine the convection and radiation constants of bright tin plate, the metal used for screens in this investigation. With the numerical values of these constants substituted in the usual heat transfer equations it is possible *first* to check the results obtained by direct test, and *second* to determine the amounts of energy transferred by conduction, convection and radiation respectively.

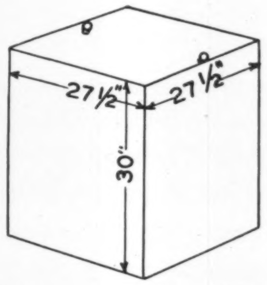


FIG. 1—ISOMETRIC VIEW OF EXTERIOR OF BOX TESTED SHOWING DIMENSIONS

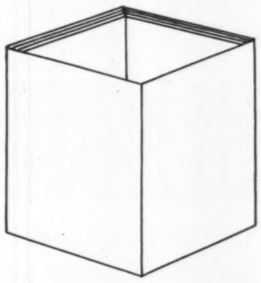


FIG. 2—ISOMETRIC VIEW OF BOX TESTED WITH LID OFF

Box Test

Figs. 1, 2 and 3 show the construction of the box, which was tested both as a refrigerator and as an oven. Inside and outside surface temperatures were obtained with thermocouples soldered to the surfaces. When tested as a refrigerator the heat transfer was determined by the rate at which ice melted inside the box, and when converted into an oven heat was supplied electrically.

To facilitate comparisons with other heat insulating structures an overall heat transfer coefficient (k_t) is introduced which may be defined by the following well known equation for the conductivity of homogeneous solids bounded by plane parallel faces kept at constant temperatures t_1 and t_2, respectively:

$$*k_t = \frac{QL}{(A)\ (time)\ (t_1 - t_2)} \qquad (1)$$

where

k_t = Btu per hour per square foot at a temperature gradient of one degree Fahrenheit per inch
Q = total quantity of heat transferred
L = length of heat path
A = area of heat path

No rigid mathematical solution has been obtained for the shape factor of cubical enclosures surrounded by walls of uniform thickness. An approximate formula developed by Langmuir, Adams and Meikle,[3] applicable to the box in this test is:

$$S = \frac{A}{L} + 0.54\ \Sigma\ D + 1.2L \qquad (2)$$

where

S = shape factor
A = inside surface area
L = wall thickness
ΣD = summation of internal edges

Using the equation

$$k_t = \frac{Q}{(t_1 - t_2)\ (time)\ S} \qquad (3)$$

for the value of k_t the following results were obtained:

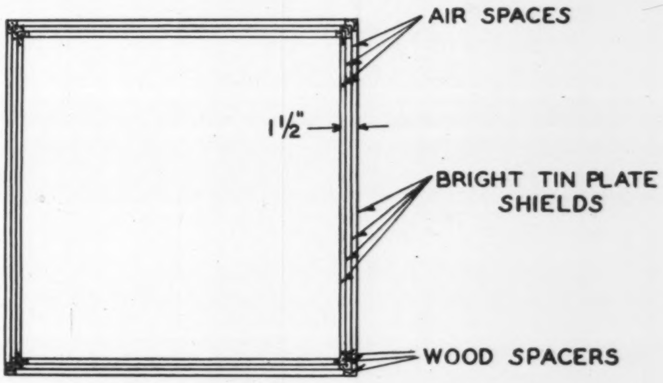


FIG. 3—SECTION THROUGH BOX TESTED, SHOWING TIN PLATE SHIELDS AND WOOD SPACERS PROVIDING FOR THE AIR SPACES

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[1] Research engineer, Westinghouse Electric and Manufacturing Company.

* As will be shown, k_t is not a constant, but is a function of t_1 and t_2. This should be taken into account when making comparisons.

[2] Warmeschutz durch Aluminiumfolie, Prof. Dr.-Ing Ernst Schmidt, V. D. I. 71, 1395, Oct., 1927.

[3] Dictionary of Applied Physics 1,463.

t_1	t_2	k_t
69 F	50 F	0.24
98	78	0.30
112	83	0.32
165	92	0.35

No great accuracy is claimed for the values of k_t given in the foregoing tabulation since it was impossible to maintain isothermal conditions on either the internal or external surfaces.

Standard Test

The set-up for the usual method of guarded hot plate sandwiched between similar samples with guard ring along the edges is shown in Figs. 4 and 5. In this case three shields of bright tin plate were used in each sample and the air spaces were again ½ in. thick. Wooden frames separated the guard ring from the test samples and closed the outside edges of the guard ring. Ten-mil mica strips turned edgewise separated the plates. The following results were thus obtained:

t_1	t_2	k_t
176 F	103 F	0.40

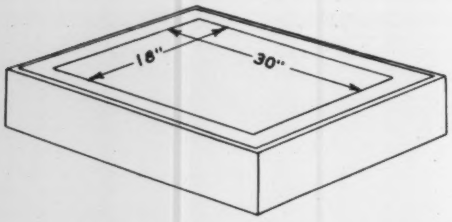


FIG. 4—PERSPECTIVE OF GUARDED HOT PLATE TEST APPARATUS

While it was possible to obtain fairly uniform temperatures over the hot and cold surfaces, there was undoubtedly some heat transmitted by conduction along the mica spacers and wooden frames so that the value of 0.4 for k_t is perhaps slightly high.

Heat Dissipation from Bright Tin Surfaces

The usual equation for heat dissipation from surfaces by natural convection and radiation is

$$q = \frac{Q}{(A)\ (time)} = k_s\ (T_1 - T_2)^d + (e)\ (0.172) \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4 \right] \qquad (4)$$

where

q = rate of heat transfer in Btu per hour per square foot

e = the radiation constant which is equal to unity for a *black body*

T_1 and T_2 are temperatures in degrees Fahrenheit absolute of the dissipating surface and surroundings respectively.

k_s and d are convection constants

The radiation and convection constants for a sheet of bright tin plate suspended in air were obtained from a curve of total heat dissipated plotted against surface temperature rise. It was necessary of course to maintain the surroundings at a fairly uniform temperature which in this work was 77 F. The numerical values of these constants were found to be:

$$k_s = 0.329$$
$$d = 1.25$$
$$e = 0.15$$

The convection constants will be modified as the screens are brought closer together, but they are applicable to air spaces ½ in. or more in thickness.[4]

The Effect of Metallic Screens

Each of the methods of heat transfer—radiation, convection and conduction—will be considered separately as though the others were for the time being inoperative. The synthesis of the separate results in obtaining an overall coefficient of heat transfer introduces some error, which, however, is negligible if the temperature difference between the hot and cold plates is not over 200 or 300 F.

1. *Radiation:*

Consider two parallel surfaces of infinite extent, A the radiator and B the receiver. Let the temperatures of the surfaces be maintained at T_1 and T_2 degrees Fahrenheit absolute, respectively. The rate of heat interchange per square foot by radiation from A to B is

$$q = 0.172 \frac{e_1 e_2}{e_1 + e_2 - e_1 e_2} \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4 \right] \qquad (5)$$

If both surfaces are alike, then $e_1 = e_2$ and

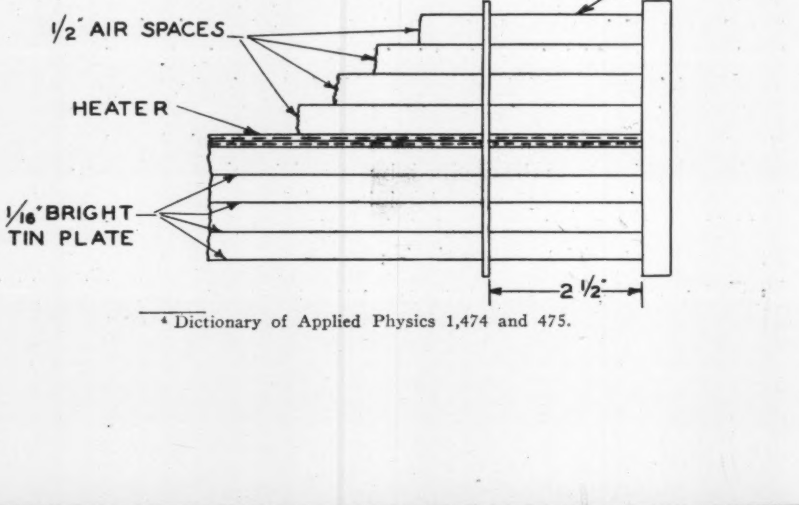
$$\frac{e_1 e_2}{e_1 + e_2 - e_1 e_2} = \frac{e^2}{2e - e^2} \qquad (6)$$

Let a certain number of screens (n) be placed between the two surfaces and parallel with them. Now the heat flow rate from A to B is less than with no screens. If all surfaces are alike the problem is simplified and the effect of the screens can be predicted.

$$Let\ a = \left(\frac{T_1}{100}\right)^4$$

$$X_1 = \left(\frac{T_{s1}}{100}\right)^4$$

FIG. 5—SECTION THROUGH GUARDED HOT PLATE APPARATUS



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[4] Dictionary of Applied Physics 1,474 and 475.

$$X_2 = \left(\frac{T_{sn}}{100}\right)^4$$

$$X_n = \left(\frac{T_{sn}}{100}\right)^4$$

$$b = \left(\frac{T_2}{100}\right)^4$$

where

T_1 = temperature of the hot or emitting surface A
T_{s1} = temperature of first screen
T_{s2} = temperature of second screen
T_{sn} = temperature of n^{th} screen
T_2 = temperature of the cool or receiving surface B

Let $\dfrac{0.172\,e^2}{2e - e^2} = E$, where E is a constant for all the screens as well as A and B if all surfaces are alike.

Let q_n = rate of heat transfer from A to B with n screens between the two surfaces. *Then*

$$\frac{q_n}{E} = a - X_1 = X_1 - X_2 = \ldots\ldots\ldots\ldots X_n - b$$

$$X_1 = a - \frac{q_n}{E}$$

$$X_2 = a - \frac{2q_n}{E}$$

$$X_n = a - \frac{nq_n}{E}$$

$$b = a - \frac{(n+1)\,q_n}{E}$$

$$q_n = \frac{(a-b)\,E}{n+1} = \frac{\dfrac{0.172\,e^2}{2e-e^2}\left[\left(\dfrac{T_1}{100}\right)^4 - \left(\dfrac{T_2}{100}\right)^4\right]}{n+1} \qquad (7)$$

That is, for n screens all having the same coefficient of emissivity as the emitting and receiving surfaces, the total radiation is $\dfrac{1}{n+1}$ times the total radiation without screens.

It is interesting to note that the temperature of any of the screens may be obtained also. Let it be required to find the temperature of the p^{th} screen where the total number of screens is n.

$$X_p = a - p\left(\frac{q_n}{E}\right)$$

but $\dfrac{q_n}{E} = \dfrac{a-b}{n+1}$

hence $X_p = a - p\left(\dfrac{a-b}{n+1}\right)$

and T degrees Fahrenheit (absolute) of the p^{th} screen is $100\sqrt[4]{a - p\left(\dfrac{a-b}{n+1}\right)}$

2. *Convection:*

Free convection in air is given by the equation

$$q = C\,(t_1 - t_2)^d \qquad (8)$$

C and d are constants to be determined experimentally.

$(t_1 - t_2)$ is the average temperature difference between the heated surface and ambient air.

The heat transfer by convection across an air space bounded by two parallel surfaces is $q = \dfrac{C}{2^d}\,(t_a - t_b)^d$ in which t_a and t_b now represent the average temperature difference between the two surfaces.

Let $k_S = \dfrac{C}{2^d}$ and the convection equation becomes

$$q = k_S\,(t_a - t_b)^d \qquad (9)$$

With n screens between the two surfaces

$$q_n = k_S\,(t_a - t_1)^d + k_S\,(t_1 - t_2)^d = \ldots\ldots K_S\,(t_n - t_b)^d$$

where q_n = rate of heat transfer between surfaces A and B with n parallel surfaces equally spaced between them.
t_a = temperature of surface A.
t_1 = temperature of first screen.
t_n = temperature of n^{th} screen.
t_b = temperature of surface B.

Let $(t_a - t_1)$, $(t_1 - t_2) \ldots\ldots (t_n - t_{n-1})$, $(t_n - t_b) = X_1, X_2 \ldots X_n$ and X_{n+1}

Let X_t = total temperature drop from surface A to surface B. Since $k_S\,X_1{}^d = k_s\,X_2{}^d$, etc.

$X_1 = X_2 = \ldots\ldots\ldots X_n = X_{n+1}$, and
$X_t = (n+1)\,X_1 = (n+1)\,X_2$, *or*

$$X_1 = \frac{X_t}{n+1}, \textit{ and}$$

$$q_n = \frac{k_s\,(X_t)^d}{(n+1)^d} \qquad (10)$$

3. *Conduction:*

With n screens per inch of heat path each screen L inches thick

$$q_n = \frac{t_1 - t_2}{\dfrac{1-nL}{k_a} + \dfrac{nL}{k_m}} \qquad (11)$$

$t_1 - t_2$ = temperature difference between hot and cold surfaces.
k_a = thermal conductivity of air.
k_m = thermal conductivity of screens.

It is apparent therefore that the amount of heat transferred by conduction is increased by the use of metal screens, but of course the thinner the screens the less this increase will be.

Summarizing and Comparing

The equation for the rate of heat transfer across an air space without screens is

$$q = \frac{(T_1 - T_2)\,k_a}{L_a} + k_s\,(T_1 - T_2)^d + \frac{0.172\,e^2}{2e - e^2}\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \qquad (12)$$

The corresponding equation for the case where n screens are used is obtained by summing equations (7), (10) and (11) and is as follows:

$$q_n = \frac{(T_1 - T_2)}{\dfrac{L_a}{k_a} + \dfrac{L_m}{k_m}} + \frac{k_s\,(T_1 - T_2)^d}{(n+1)^d} + \frac{\dfrac{0.172e^2}{2e - e^2}\left[\left(\dfrac{T_1}{100}\right)^4 - \left(\dfrac{T_2}{100}\right)^4\right]}{n+1} \qquad (13)$$

In equations (12) and (13)

q = rate of heat transfer without screens.

q_n = rate of heat transfer with n screens.

T_1 and T_2 = temperatures in degrees Fahrenheit (absolute) of hot and cold surfaces.

L_a = length of heat path through air.

L_m = length of heat path through metal screens.

k_a = thermal conductivity of air.

k_m = thermal conductivity of metal used for screens.

k_s and d = convection constants, determined experimentally.

e = emissivity of metal used in per cent of black body radiation.

Heat Transfer Through Air Spaces

The value of air spaces as thermal insulation is often discussed without reference to the character of inclosing walls. The character of bounding surfaces greatly affects the amount of energy transferred by radiation which is the only way of modifying the total heat transferred across a given air space. The importance of radiation is often overlooked, especially in problems involving ordinary room temperatures. To show the relative effects of radiation, convection and conduction it is only necessary to evaluate equations (12) and (13) with the numerical values of the various constants inserted.

The following examples will illustrate the use of these equations and at the same time show how much the transfer across a given air space may be modified. The distance between the hot and cold walls is 1½ in. and the temperatures of the hot and cold surfaces are 212 F and 32 F, respectively. In *Case 1* the inclosing walls are paper, wood, asbestos or other similar material having a radiation constant of approximately 0.9, while in *Case 2* the walls are bright metal with a radiation coefficient of 0.15. In *Case 3* two bright metal screens are used thus dividing the inclosure into three ½-in. spaces.

Case 1

k_a, or air conductivity at 122 F is 0.174

k_s, convection constant = 0.141

d, convection constant = 1.25

$\dfrac{e^2}{2e - e^2}$ radiation constant = 0.82

Substituting these values in equation (12)

$$q = \frac{(0.174)\ (180)}{1.5} + 0.141\ (180)^{1.25} + (0.172)\ (0.82)\ (6.72)^4 - 4.92^4$$

$*q = 21 + 92 + 206 = 319$

The heat flow in Btu per square foot per hour is therefore as follows:

21 Btu or 6.6 per cent of the total is transferred by conduction.

92 Btu or 28.9 per cent of the total is transferred by convection.

206 Btu or 64.5 per cent of the total is transferred by radiation.

319 total Btu per square foot per hour.

The overall transfer coefficient is

$$k_t = \frac{(319)\ (1.5)}{180} = 2.66$$

Case 2

The only change in *Case 2* is the radiation coefficient which instead of 0.82 as for *Case 1* is 0.081. Substituting and solving as before the following results are obtained:

21 Btu or 15.8 per cent of the total is transferred by conduction.

92 Btu or 69.1 per cent of the total is transferred by convection.

20 Btu or 15.1 per cent of the total is transferred by radiation.

133 total Btu per square foot per hour.

The overall coefficinet of heat transfer is

$$k_t = \frac{(133)\ (1.5)}{180} = 1.11$$

By replacing the surfaces in *Case 1* with bright metal the heat flow has been reduced to less than one-half of the previous value. This was accomplished entirely by reducing the radiation, without changing the convection and conduction.

Case 3

As already stated, the conditions for *Case 3* are the same as for *Case 2* except that two bright metal screens now divide the inclosure into three ½-in. spaces. Solving equation (13) with the numerical values of the constants substituted and assuming a screen thickness of 1/16 in. the following is obtained:

$$q_n = \frac{180}{\dfrac{1.5 + 2\,(1/16)}{0.174} + \dfrac{0.125}{273}} + \frac{\text{previous convection}}{(2+1)^d} + \frac{\text{radiation of } \textit{Case 2}}{2+1}$$

$= 23 + 23 + 7 = 53$ which means that in one hour and across one square foot of surface

23 Btu or 43.4 per cent of the total is transferred by conduction

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* Computations by slide rule and values given to nearest whole number.

23 Btu or 43.4 per cent of the total is transferred by convection
7 Btu or 13.2 per cent of the total is transferred by radiation
—
53 total Btu per square foot per hour.

The overall coefficient of heat transfer is

$$k_t = \frac{(53)\ (1.5)}{180} = 0.44$$

Discussion of Results

The coefficient of heat transfer as derived from the direct tests averages about 0.38 at an average temperature of 135 F while the corresponding value obtained by the indirect method is 0.44 which is quite satisfactory agreement in view of the uncertainties involved.

As stated previously some error is introduced by considering the three different ways of heat transfer separately in obtaining the rather *synthetic* coefficient. The writer knows no method of obtaining an expression for the heat transfer coefficient through successive air spaces by using the radiation, convection and conduction constants simultaneously except by trial and error which would also be more or less approximate and very much more laborious.

Summary

The rate of heat flow between two parallel surfaces, *A* and *B*, in air separated a distance *L* from each other is given by the equation

$$q = \frac{k_a (T_1 - T_2)}{L} + k_s \ (T_1 - T_2)^d + \frac{(e_1 e_2) S}{e_1 + e_2 - e_1 e_2} \left[(T_1)^4 - (T_2)^4 \right] \qquad (14)$$

where

k_a = thermal conductivity of air
k_s and d = convection constants
e_1 and e_2 = the radiation constants of the respective surfaces
S = black body emissivity = 0.172×10^{-8}
T_1 and T_2 = the absolute temperatures of the respective surfaces
q = Btu per square foot per hour
k_S = approximately 0.14 for most smooth flat surfaces
d = 1.25
e = approximately 0.9 for wood, paper, asbestos, etc., and from 0.05 to 0.20 for bright metallic surfaces.

Where bright metal screens are used between two surfaces of like material, that is for the case of successive air spaces between parallel surfaces of bright metal, the rate of heat transfer is given by the equation

$$q_n = \frac{T_1 - T_2}{\dfrac{L_a}{K_a} + \dfrac{nL_m}{K_m}} + \frac{k_S (T_1 - T_2)^d}{(n+1)^d} + \frac{\dfrac{e^2 S}{2e - e^2}(T_1^4 - T_2^4)}{n+1}$$

where

q_n = rate of heat transfer from surface *A* to surface *B* with *n* screens dividing the inclosed space from *A* to *B* in $n + 1$ equal parts.
L_a = length of heat flow path in air.
k_a = thermal conductivity of air.
L_m = thickness of metal screen.
k_m = thermal conductivity of air.

With the proper value of the constants substituted in the foregoing equations it is not only possible to determine q and q_n but also the amounts of heat transferred by and therefore the relative importance of conduction, convection and radiation in problems of heat flow across air spaces.

Three examples involving the use of the foregoing equations have been given which are briefly summarized as follows:

Case 4.

Air space without screens between two good radiating surfaces such as cork, paper, wood, etc. Let $e = 0.9$, which is approximately correct for such materials.

Case 5.

Air space without screens between two poor radiating surfaces such as bright metals where $e = 0.15$.

Case 6.

Same conditions as for *Case* 5, with the exception that two bright tin plates divide the 1½-in. air space into three ½-in. spaces. Common to all three cases: distance from the hot to the cold surfaces is 1½ in. and the respective temperatures are 212 F and 32 F.

A summary of the results is given in Tables 1 and 2.

TABLE 1—BTU TRANSFERRED PER HOUR PER SQUARE FOOT BY CONDUCTION, CONVECTION AND RADIATION

	CASE 1	CASE 2	CASE 3
Conduction	21	21	23
Convection	92	92	23
Radiation	206	20	7
Total	319	133	53

TABLE 2—PERCENTAGES OF TOTAL HEAT TRANSFERRED BY CONDUCTION, CONVECTION AND RADIATION AND VALUES OF OVERALL HEAT TRANSFER COEFFICIENT

	CASE 1	CASE 2	CASE 3
Conduction	6.6%	15.8%	43.4%
Convection	28.8%	69.1%	43.4%
Radiation	64.5%	15.1%	13.2%
Coefficient (k_t)	2.66	1.11	0.44

Effect of Bright Metallic Screens on Conduction, Convection and Radiation

The transfer by conduction is increased, the amount of increase depending upon the extent to which the air path, a poor conductor, is displaced by metal, which is a much better conductor. Convection is decreased as follows: The heat transfer by convection is given by

$q = K_S\ (t_1 - t_2)^d$ without screens.

$q = \dfrac{K_S\ (t_1 - t_2)^d}{(n+1)^d}$ with n screens.

Radiation is decreased, the amount of decrease being conveniently shown as follows:

Let q = transfer by radiation without screens.

Then $\dfrac{q}{n+1}$ = transfer by radiation with n screens.

Experiments

Two direct tests were made to determine the heat transfer coefficients of successive half-inch air spaces separated by bright tin plate.

Standard Test

The usual method of guarded hot plate sandwiched between two similar samples with guard ring along the edges gave a transfer coefficient of 0.40 Btu per square foot per hour per degree Fahrenheit per inch with t_1 and t_2 at 176 F and 103 F, respectively.

Box Test

A box having inside dimensions of 24 × 24 × 26 in. was tested both as a refrigerator and as an oven. The walls of the box were 1.75 in. thick with three ½-in. air spaces separated and inclosed by 1/16-in. bright tin plate. The following results were obtained:

t_1	t_2	k_t
69 F	50 F	0.24
98	78	0.30
112	83	0.32
165	92	0.35

New Ventilation Code for New York City

The ventilation problem has been attacked in an entirely new manner in the new Building Code which The Merchants' Association of New York is now writing at the request of the City Administration. The proposed provisions of the new code are contained in a report of the Code Committee's Subcommittee on Heating and Ventilating. This subcommittee consists of Clyde R. Place, B. H. Belknap, W. H. Driscoll, Charles Hoffman, Arthur K. Ohmes and Frederic N. Whitley.

In general, the proposed code, as drawn by the subcommittee, intends that many spaces in which the air is now *dead* shall be ventilated by artificial means. The proposed code provides that in all cases the ventilation shall be sufficient to render the air harmless for the occupancies that are permitted.

As a means of accomplishing its purpose of providing pure air for everyone, the committee has recommended the establishment of a simple basic formula. Under this formula an *index figure* based on window space, floor area, cubical contents and proposed occupancy is established for all types of buildings.

The present New York building code attempts to deal with the ventilation problem by permitting the presence of not more than one part of carbon dioxide to a thousand parts of air. It is claimed that this test has been impracticable because it is almost impossible to determine the amount of carbon dioxide that will be present under varying conditions. It has been found that even when the carbon dioxide content is below the specified amount, areas may still be badly ventilated. In the judgment of the committee the carbon dioxide index has been worthless.

The proposed code, as written by the subcommittee, reads as follows:

"Spaces above grade, with or without windows, designed for human occupancy only, shall have ventilation either from windows or from mechanical means, or from both, in accordance with the following index and requirements:

"Cubic contents per person plus 10 times floor area per person plus 100 times window opening per person equals index.

"For Rooms with Windows

"If the Index is less than 300 there shall be supplied an amount of fresh air equal to 2½ cu ft per minute per square foot of floor area, and an air exhaust of 2 cu ft per minute per square foot of floor area.

"If the Index is between 300 and 520 there shall be supplied an amount of fresh air equal to 2 cu ft per minute per square foot of floor area, and an air exhaust of 1½ cu ft per minute per square foot of floor area.

"If the Index is between 520 and 850 there shall be supplied an amount of fresh air equal to 1½ cu ft per minute per square foot of floor area and an air exhaust of 1¼ cu ft per minute per square foot of floor area.

"If the Index is between 850 and 1,650 there shall be required an air exhaust only of one cubic foot per minute per square foot of floor area.

"If the Index is above 1,650 no ventilation is required.

"For Rooms without Windows

"If the Index is below 850 the requirements shall be the same as for rooms with windows.

"If the Index is between 850 and 1,650 there shall be required an air supply of one cubic foot per minute per square foot of floor area and an air exhaust of one cubic foot per minute per square foot of floor area.

"If the Index is over 1,650 there shall be required an air supply of 1/3 cubic foot per minute per square foot of floor area and an air exhaust of 1/3 cubic foot per minute per square foot of floor area.

"Interior partitions shall have transoms, and when partitions occur 30 ft or more away from a window or similar opening, the room so formed shall have ventilation based upon the Index without windows.

"Plans for structures except one and two family dwellings, designed for human occupancy and filed with the Building Department, shall have designated thereon the number of persons which the rooms and various spaces are planned to accommodate.

"Spaces below grade without windows and designed for human occupancy shall have a mechanical means of ventilation of at least 50 cu ft per person per minute and at least four changes per hour."

Subcommittee No. 3 of the *American Standards Association* on Steel Pipe Flanges and Flanged Fittings of the Sectional Committee on the Standardization of Pipe Flanges and Fittings, has recently released for general distribution a proposed American standard for steel base fittings, which is now being widely distributed to industry in draft form for criticism and comment.

This proposal originated with the Manufacturers Standardization Society of the valve and fittings industry by whom it was adopted as *standard practice* in October, 1929. This standard is not complete in itself but must be treated as a supplement to, and used in conjunction with, the American standard for steel pipe flanges and flanged fittings.

Off-Peak System of Electric Heating for Buildings

By Elliott Harrington[1], Schenectady, N. Y.
MEMBER

MANY years ago, Dr. Charles P. Steinmetz predicted that some day homes would be built without chimneys and heated entirely by electricity. Today, the average informed man knows that he can heat his home electrically if he chooses to do so, but he immediately associates exorbitant cost with this operation. He knows that electric heat would be clean, safe, convenient, silent, and reliable. But his knowledge usually ends with the 500-watt radiant heater which he sets in his bathroom on a cold morning. He has learned that when he uses this radiant heater consistently he receives a materially increased electric power bill at the end of the month and he has only electrically heated one small room a part of the time. However, the prophecy of Dr. Steinmetz is much nearer an economic and practical realization today than is generally realized, and off-peak power appears to be the solution of the problem.

During most of the daytime and evening when the electrical consumers are drawing power at or near the full capacity of the power station, any increase in load immediately requires an increased investment in generating equipment. However, during certain hours of the twenty-four, the demand on the generating equipment is very low. The particular hours will vary, of course, for various localities, depending upon the characteristic use of energy in that particular territory. These hours of small usage, or *off-peak* hours, as they are known, in practically all territories include the hours from 10:30 p. m. to 7:00 or 8:00 a. m., and in most territories include certain periods during the day. These hours of usage are very firmly established by the type and habits of the electrical customers, and it is possible to adapt the use of apparatus to the *off-peak* hours characteristic of any particular locality. The power generated during these off-peak hours involves costs for fuel, incidental labor, and supplies, but does not involve additional investment cost for generating equipment.

Hence, as the telegraph company is ready to send a day letter or a night letter at a reduced rate when the wires are not busy, so the central station is ready to supply electric energy at a low rate when the apparatus which they already have installed is relatively idle. The problem of electric heating of buildings, therefore, becomes one of storing this electric energy purchased during the low rate period and using this stored energy as required.

A number of different methods have been tried for storing this electrical energy and using it later in the heating of buildings. The original and most obvious method was an effort to store and use the electricity itself by charging storage batteries, but the first cost of the apparatus for any reasonable storage capacity was prohibitive.

Then media were tried for storing this energy in the form of electric power converted into heat. Sand, crushed rock, cast iron, soapstone and water have all been used with varying degrees of success. Probably water is the most acceptable because it is universally available, costs nothing and, since it is a fluid, it can be circulated. It is immediately adaptable to the conventional hot water radiation system.

Cost

Before discussing the apparatus in detail and its application to various heating problems, consider the question of paramount importance, namely the cost of operating.

Of course, the first cost of the installation varies with the characteristics of the building. However, the installation cost of a typical residence of 22,000 to 28,000 cu ft is about comparable to the cost of a high grade boiler, oil burner, control, and a 1,000-gal outside oil tank. A complete gas heating system, consisting of gas boiler, burner, and control, would cost about two-thirds as much as the electric system, and a complete coal burning system about one-third as much. In a new building, the first cost of the radiation can be reduced somewhat with electric heating because the higher temperature of the circulating water requires less standing radiation, and with forced circulation the distributing pipes can be smaller.

The operating cost depends upon a number of variables, among which are the power rate obtained from the central station and the characteristics of the building to be heated. Central stations are rapidly accepting electric heating as a thoroughly workable means of increasing their load factor (ratio of average load to maximum load demand). In many states—notably Florida, Georgia, Illinois, Indiana, Iowa, Ohio, Virginia, Pennsylvania and New Jersey—*off-peak* domestic water heating and house heating rates have been established ranging from approximately 0.75c per kilowatt hour to 1.5c per kilowatt hour.

Since the operating cost is dependent upon so many variables, a broad and accurate statement of this cost cannot be made. Generally speaking, a rate of 1c per kilowatt hour will give an operating cost approximately equal to the cost of heating with manufactured gas at $1.10 per thousand cubic feet. A rate of ½c per kilowatt hour would give a cost approximately equal to that of anthracite coal at $18.00 per ton.

During the winter of 1928-1929, seventeen installations of domestic electric heating were made in Ft.

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TABLE 1—ENERGY USED FOR ELECTRICALLY HEATING EIGHTEEN HOUSES IN FORT WAYNE, IND.
Oct. 15, 1929, *to May* 15, 1930

HOUSE	DESCRIPTION OF HOUSE	NEW OR OLD HOUSE	INSULATION	CUBICAL CONTENT HEATED	KILOWATTS INSTALLED	NET KILOWATT-HOURS USED[a]
1	Brick veneer	Old	Walls and ceiling	44,840	80	122,310
2	Brick veneer	Old	Ceiling and double glass	32,499	60	85,970
3	Brick veneer	New	Walls, ceiling and double glass	28,255	50	35,695
4	Clapped frame	Old	Walls, ceiling and double glass	28,124	50	78,750
5	Stucco frame	Old	Walls, ceiling and double glass	26,250	60	70,615
6	Brick veneer	New	Walls and ceiling	22,650	40	35,833
7	Brick veneer	New	Walls and ceiling	22,400	60	51,443
8	Brick veneer	New	Walls and ceiling	21,920	60	55,146
9[b]	Stone	Old	Ceiling and double glass	21,277	60	68,817
10	Clapped frame	Old	None	20,020	50	60,690
11	Clapped frame	Old	None	20,020	50	54,922
12	Clapped frame	Old	Walls and ceiling	15,700	40	37,800
13	Clapped frame	Old	Walls and ceiling	15,620	50	40,025
14	Brick veneer	New	Walls, ceiling and double glass	14,000	30	25,222
15	Shingle frame	New	Walls, ceiling and double glass	13,690	30	13,900
16	Brick veneer	New	Walls, ceiling and double glass	13,170	30	19,080
17	Clapped frame	New	Walls, ceiling and double glass	12,420	30	14,990
18	Shingle frame	New	Walls, ceiling and double glass	9,130	30	17,170

[a] Some variation between size of house and kilowatt-hour consumption may be accounted for by house temperatures maintained by individual householders, and other unusual operating conditions.
[b] House No. 9 located in Philadelphia, Pa.; all others in Ft. Wayne, Ind.

Wayne, Indiana. The operating cost of heating these seventeen homes ranged from $190.00 for a 9,130 cu ft home to $1,300.00 for a 44,840 cu ft home (see Tables 1 and 2 and Fig. 1). For these installations, off-peak power was purchased at the rate of 2c per kilowatt hour for the first 400 kw hours used per month and 1c per kilowatt hour thereafter.

In considering the operating cost of electric heating, it must be remembered that in this system electricity is

TABLE 2—AVERAGE TEMPERATURES DURING JANUARY AND FEBRUARY IN FORT WAYNE, INDIANA, FROM 1925 TO 1929
Based on records of Ft. Wayne branch of U. S. Weather Bureau

YEAR	TEMPERATURE (DEGREES F)	
	JANUARY	FEBRUARY
1925	23.6	34.4
1926	27.1	30.4
1927	25.2	35.2
1928	26.8	29.7
1929	21.4	23.1

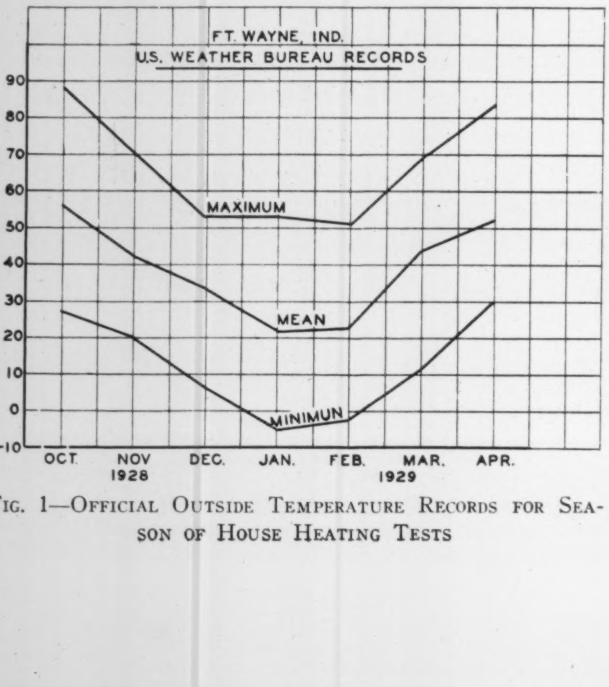

FIG. 1—OFFICIAL OUTSIDE TEMPERATURE RECORDS FOR SEASON OF HOUSE HEATING TESTS

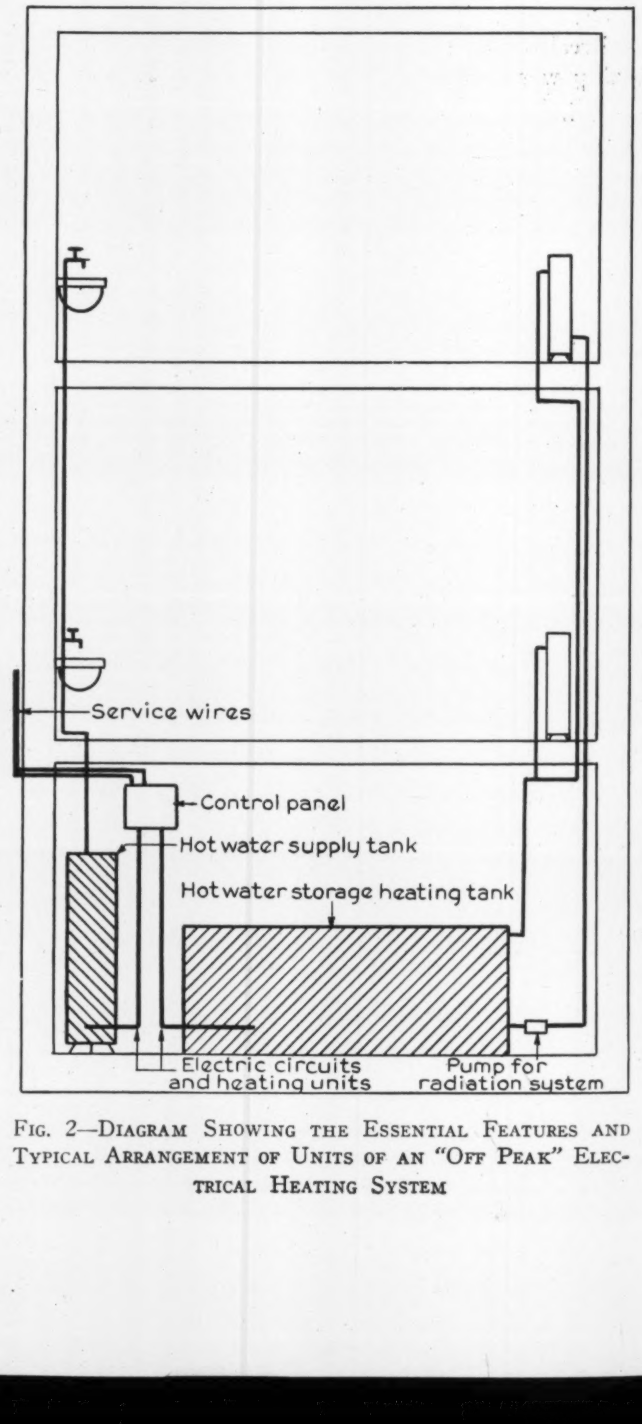
Service wires
Control panel
Hot water supply tank
Hot water storage heating tank
Electric circuits and heating units
Pump for radiation system

FIG. 2—DIAGRAM SHOWING THE ESSENTIAL FEATURES AND TYPICAL ARRANGEMENT OF UNITS OF AN "OFF PEAK" ELECTRICAL HEATING SYSTEM

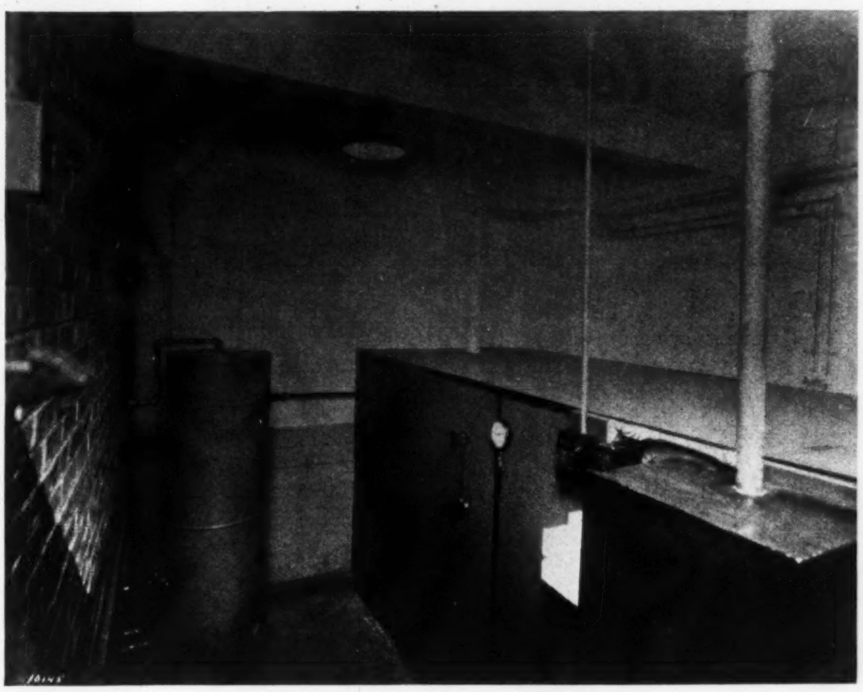

FIG. 3—A COMPLETE HOT WATER UNIT INSTALLED IN A SPACE 6 FT BY 10 FT; UNITS FOR SMALL HOMES CAN BE INSTALLED IN 24 SQ FT OF FLOOR SPACE

converted into heat at an efficiency of 100 per cent. Every kilowatt-hour provides 3,415 Btu of heat energy and every one of these Btu is released into the home. Therefore, the electric meter immediately becomes a Btu meter. Furthermore, it is not necessary to draw any air from out of doors, heat it to room temperature, use it to support combustion, and then dispose of it through the chimney.

Advantages

The advantages of heating electrically are generally known, but it is well to review them briefly as follows:

1. *Safety.* As safe as the electric wiring system now installed in 19,000,000 homes, and it operates without flame or combustion of any kind. No combustible is brought into or stored in the house.
2. *Reliability.* There has never been an electric power famine, and prolonged power failure in a given locality is almost unknown.
3. *Cleanliness.* No solid, liquid or gaseous fuel comes into the building, and no ashes go out. No smoke, fumes or soot are released in or around the building.
4. *Quietness.* The only moving mechanism is the heat circulating device which can be made to operate without noise.
5. *Availability.* The electric heating unit, because it does not involve open flame, and because the storage tank can be made of different shapes—that is, tall and vertical or broad and horizontal—can be placed in almost any desired part of a dwelling or other building.
6. *Low Maintenance.* Absence of combustion tends to reduce maintenance to a minimum.

Description

Briefly, the apparatus for the electric heating of buildings using a hot water system, consists of the following (see Figs. 2, 3, 4 and 5):

1. Heat insulated tank for the storage of hot water.
2. Electric immersion units for heating the water in the tank.
3. Control panel containing main switch, time and temperature control switches.
4. Circulating pump (for warm air systems, a fan circulating air around the heat storage tank and through the hot air ducts would be substituted for the circulation pump).

Operation

The operation of the system is as follows: During the *off-peak* period an electrically driven time switch energizes the immersion units which heat the water in the storage tank to 250 F (for warm air systems the water is heated to 300 F). When the predetermined temperature is reached, a pressure switch cuts off the power with enough heat stored in the tank to heat the building in question for 24 hours under the most severe weather conditions. When the room thermostat calls for heat, it starts the circulating pump which circulates hot water from the storage tank to the radiators. With warm air, the fan is started, thereby circulating air through a jacket around the hot water storage tank or through a unit heater to the registers. When proper room temperature is obtained, the thermostat shuts down the circulating unit. The heat storage tank is so well insulated that the heat loss from it is very low. The small amount of heat given off by the heat storage warms the basement or whatever part of the building houses the storage tank.

Of course, heat insulation of the building is an important part of any heating problem and it is particu-

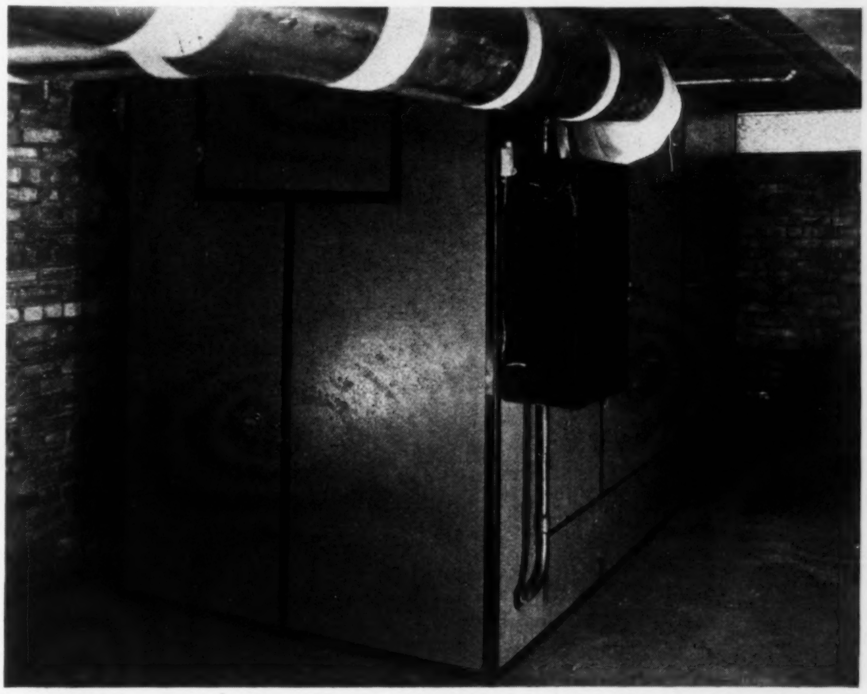

FIG. 4—A TYPICAL WARM AIR ELECTRIC HEATING UNIT

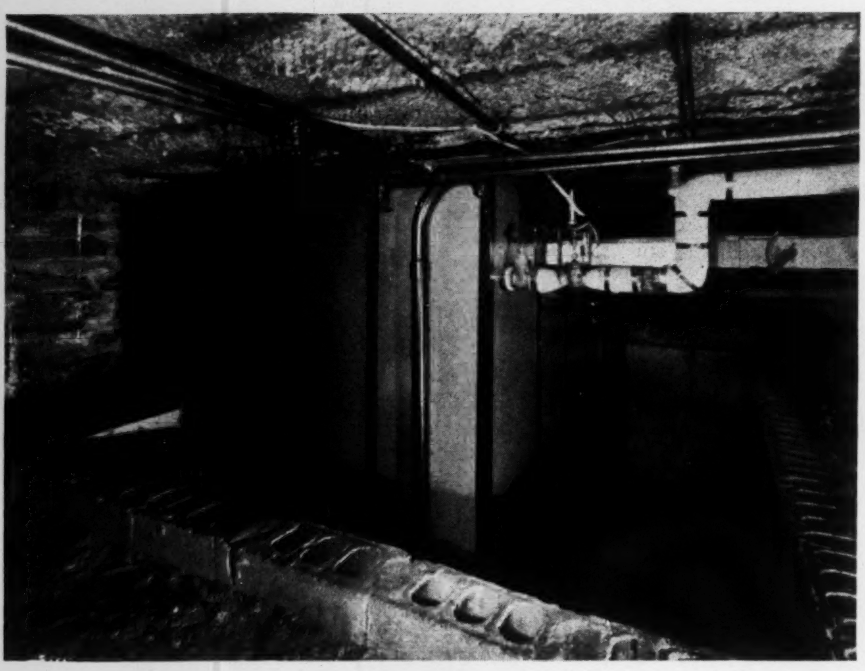

Fig. 5—A Hot Water Electric Heating Unit Installed in a Pit and Without a Basement

larly important with electric heat. It is a well recognized fact that the installation of the proper thickness of heat insulation, together with double windows and weather strips, will reduce the heat losses from the building about 50 per cent.

Domestic Water Heating

An *off-peak* water heater has been developed as a part of this system and can well be considered with it. This water heater (Fig. 6) is particularly important from the standpoint of the central station, since to improve the load factor during the summer months, the *off-peak* period must be utilized to maintain the low power rate throughout the year. This *off-peak* water heater operates on the same principle as the system for building heating—that is, during the *off-peak* period, water is heated in an insulated storage tank to a temperature of about 200 F. The water is then drawn from this tank as needed during the day through a thermostatically controlled mixing valve which mixes sufficient cold water with the water from the heat storage tank to produce the desired temperature of perhaps 130 to 140 F. When this water heater is installed with the electric heating equipment for the building, the same time switch and electric meter controls both units.

Application

In the past three seasons approximately fifty complete installations of electric heating systems have been made in various parts of the United States and have operated successfully over the range of climatic conditions existing.

In addition to the heating of homes, a number of installations have been made in sub-stations, and commercial type public utility buildings. It is significant to note that power companies in New Jersey, Ohio, Indiana, and Virginia are heating their own office buildings and substations economically with the same energy which they generate, and other power companies are now making similar installations.

This thermal storage form of electric heating possesses characteristics of flexibility and safety which will meet conditions encountered in present day buildings. As an illustration, the telephone company having small, unattended buildings largely filled with delicate automatic apparatus, which has to be maintained at a reasonably uniform temperature, must have apparatus which is safe from the standpoint of fire hazard, which is free from the products of combustion, so far as the air within the heated space is concerned, and must have an automatic form of heat which can be left for hours or days in an unattended building. Furthermore, these heated spaces may be of a few hundred cubic feet in volume to several thousand cubic feet. The thermal storage form of electric heating meets all of these conditions. Storage tanks from those of possibly two to three hundred gallons to several thousand gallons capacity can be supplied to meet the demands of the building. The source of heat—that is, electricity—is enclosed in approved iron conduit protected with approved switches and service cut-outs to meet the requirements of the *National Board of Fire Underwriters*. The automatic controls for regulating the heat can be set to maintain the temperature desired, and there is no combustion in the building. Therefore, the entire equipment meets, in every respect, the rather rigorous demands of this particular kind of an installation. This example illustrates the conditions which can be met.

Similarly, this form of electric heating undoubtedly will have a very definite effect on the architecture of homes in the future, because of its inherent characteristics of safety, freedom from combustion, and flexibility in its possibilities of installation. It is entirely possible to build a home without a cellar, which is now necessary, largely for housing a heating plant and storing fuel, and without a flue for taking away the products of combustion from the present types of heating plant located in the cellar. Inasmuch as the storage tanks for this heating system can be made of different shapes and capacities, they can be built into the structure of the house in such a manner as to eliminate the present exaggerated importance of the heating plant location in a house. A heating plant which consists essentially of a large-sized thermos bottle of varying shape, can be built into the first floor, and its electrical controls placed in the kitchen or laundry.

An apartment building of steel will be built shortly in which the amount of space required by walls, conduit, ducts, and piping will be reduced to a minimum, and each floor will supply its own heat, ventilation, etc. An electric conduit and a cold water pipe will be run from the ground floor to the top of the building, and a small *off-peak* electric heating unit will be installed in waste space under the stairs on each floor to heat that particular floor. This type of heating is particularly adaptable to a building built over a railroad right-of-way where there is no cellar and the basement becomes the first floor, which is the most valuable renting space in the entire building.

A possible field for storage-type electric heating is on the electrified railroads which buy their power on a *demand* basis. This form of contract carries a charge or penalty for the power taken during certain restricted hours, such as 7:00 to 9:00 A. M. and 5:00 to 7:00 P. M., at times when the commuting service makes the maximum draft on the electrical lines. This also coincides with the draft on the electrical lines from other types of customers. However, the amount of apparatus required is predicated on the maximum draft coming during the periods of peak load. At hours other than the demand periods, energy is generally available under this form of contract at extremely low rates. This means, therefore, that buildings along the railroad right-of-way requiring heat can be supplied with storage-type equipment utilizing cheap energy at hours other than the demand periods, and the energy can be taken directly from the lines running along the right-of-way.

An electrically driven ship is now under construction where the heating will be done electrically, using the Diesel electric propulsion equipment to charge the heat storage system when the boat is not under way.

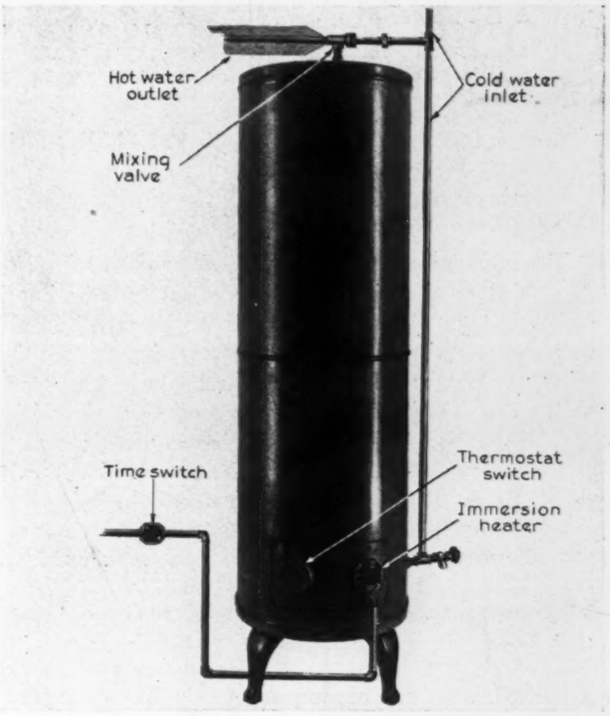

FIG. 6—OFF PEAK WATER HEATER

Conclusion

Under present conditions, and considering the load curves that are characteristic of many electric properties, there is adequate *off-peak* power available to supply from two to five per cent of the homes in any territory. The continued sale of additional lighting appliances and the many other convenient motor-driven and heating appliances for the home all tend to increase the peak demand and, correspondingly, make more *off-peak* energy available. This means that from year to year, with the present steadily increasing use of all forms of electric service, there will be made available an additional *off-peak* capacity which can be used for this form of home heating. This can be considered as a form of heating providing a maximum of safety and the utmost of convenience and cleanliness. It can be installed in place of any other form of heat and in many places where other forms of heat cannot be used.

Notes on Electric Heating

The use of electricity as a direct heating medium, employing so-called space or radiant heaters, is confined largely to mild climates, or to localities where electric rates are low. Direct electrical heaters take energy coincidentally with the heater requirements, which means that the load usually comes directly on the power station's peaks. The thermal storage system uses off-peak electrical energy for which commercially economical rates are now being made available. Therefore, this off-peak form of electrical heating is becoming practicable in many northern climates as well as in milder climates. In some cases electricity is used as a supplementary means of heating such as in industrial plants for the heating of remote spaces.

The various types may be classified as follows:

1. Direct Heaters.

a. *Radiant Type*. For quick, intermittent heating. The heating element operates at a relatively high temperature and the heat transfer is principally by radiation.

b. *Convection Type*. For continuous heating. The heating element operates at a relatively low temperature and the heat transfer is mainly by air currents.

c. *Contact Type*. Electric strip heaters placed in direct contact by clamping to objects to be heated. They may be used to prevent freezing of pipes and tanks.

2. Indirect Heaters.

a. *Hot Water or Steam Radiators*. Electric immersion units (1) to heat water in radiators, or (2) to generate steam. Each radiator is a separate unit connected to the electric wiring system.

b. *Central Heating Plant Using Instantaneous Heating*. (1) Air heated by passing directly over electric heating elements, and then circulated by a duct system to the spaces to be heated. (2) Steam generated in electric boiler at times and in quantities required by the heating demand. (3) Water heated to the temperatures and in quantities required for the heating demand.

c. *Central Heating Plant Using Thermal Storage*. This is described in the foregoing paper by Mr. Harrington and is applicable to warm air, hot water and steam heating.

The efficiency of electricity in direct heaters is 100 per cent. There is only a small loss due to transmission through the wiring system. The effective use of insulation on heat-storing water tanks as used in indirect electrical heating reduces the radiation losses practically to a negligible amount. The efficiency of electricity as a heating agent, then, is a maximum.

The more costly the fuel, the greater the need for insulation, weatherstripping and storm sash. Hence the importance of insulation and weatherstripping for **electrically heated houses cannot be overemphasized.**

Noise Reduction in Ventilating Units

By Warren Ewald[1], Philadelphia, Pa.
MEMBER

NOISE reduction is probably the most difficult and and elusive problem in the unit ventilating industry. Although there is an immense fund of knowledge covering the theory of sound in general, there has been comparatively little research work done on the problem of reducing noise, the complexity of which is increased by the difficulty of evaluating the results of experiments in terms of the comparative disagreeableness of the noises produced. This paper includes a discussion of the problem as well as the results of some research which may be useful to those concerned with noise reduction in ventilating equipment.

The problem of noise reduction apparently involves a multitude of partial solutions. The first step is to prevent, as far as possible, the creation of noises. The next is to absorb or otherwise render inaudible that portion of the noise which could not be prevented.

Sources of Sound

Sound is created in unit ventilators from the following six major causes:

1. The mechanical vibration of the rotating parts of the fans and the motor.
2. The hum in the transformer and motor windings.
3. The turbulence set up in the air during its passage through the unit.
4. The air vibration due to the action of the fan blades on the air.
5. The noise created by the mechanical movement of operating dampers.
6. Noises originating in the valves, radiator and traps.

Because of these several and separate sources, the resulting pulsation of the atmosphere is a most complex combination of widely varying intensities and wave frequencies. A large portion of the mechanical vibration of the rotating parts may be prevented by proper balancing, both statically and dynamically. Spring mountings between the motor and the motor base, minimum areas of contact between the motor and the ventilating machine and the use of various insulating materials will cause the vibration to be dissipated, although not before some portion of the vibration has been transferred to the air in the form of sound.

The second source of noise—motor and transformer hum—is caused by hysteresis and magnetic leakage. The quantity of noise from these sources is affected by the manner in which the windings are placed on the transformer core, and the manner in which the transformer is mounted. In general, a large air gap between the stator and the rotor will make a quiet motor, while a small air gap will produce greater noise. It may be necessary to sacrifice in some measure other desirable qualities such as power, compactness and efficiency, to obtain absence of noise.

There are critical velocities for all fluids below which the fluid will flow through a uniform duct without turbulence. Unfortunately these velocities are too low for most ventilating uses and it is therefore necessary to deal with air in a turbulent condition. There are, however, various degrees of turbulence, and the quantity of sound resulting from turbulence is a function of this degree. To prevent noise from turbulence, abrupt changes should be eliminated in area and direction, obstacles should be streamlined, and smooth surfaces should be provided.

The sound created by the action of the fan blades on the air cannot be entirely prevented by any means at present known. A low fan peripheral speed will reduce the noise, but only at the cost of reducing the possible air delivery from the fan. There has been some effort to prevent this noise by substituting a fan blade of curved streamline section for the present uniform thickness blade, although the results apparently have not been entirely satisfactory so far.

Dampers may be silenced by muffling the contact edges with felt, asbestos cloth or similar materials. Hinges and bearings should be such that comparative silence from these sources is assured in spite of lack of lubrication. Ordinary precautions as to proper draining will prevent any noises from condensate in the valves, traps and radiator. The radiator should have sufficient pitch, the amount of which depends on the cross sectional areas of the steam enclosure spaces and the steam velocities in these spaces.

Sound Absorption

Perhaps the best known method of reducing noise is by the absorption of the sound waves with some porous material. The interstices in the absorbing material are filled with air and present to the air passing over them, the open ends of a multitude of minute channels within which dissipation due to friction is very high. In order to prevent reflection an absorbing material must have approximately the same acoustic resistance as that of the adjacent air, and the pores and crevices must have a maximum of acoustical impedence and friction to extinguish the transmitted sound waves by conversion of the sound energy into heat. A dense, smooth-finish felt has a maximum of internal acoustic friction, but it has too much inertia, its acoustic resistance is too great, and therefore the sound waves are reflected rather than absorbed.

A series of tests were conducted to determine the proper density of felt to use as an absorbing material in unit ventilators. These tests indicated that felt was not suitable for this application. A large portion of the sound was absorbed by coarse, loose felt, but only at a sacrifice of the rate of air delivery, due to the large amount of friction of the air stream in contact with the rough felt. The loss from friction was acceptably low with a smooth, dense felt, but the sound absorption was also low. A medium grade felt proved unsatisfactory, although this material is a useful absorbing material in many applications, particularly where the air velocities are low.

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[1] Engineer, John J. Nesbitt, Inc.

Felt, coated with a thin, smooth, flexible membrane has been suggested for the purpose of absorbing sound without interfering with the flow of the air. The purpose of the flexible skin is to present to the air a smooth, low friction surface which at the same time will be thin and flexible enough so that sound impulses will be transmitted through it to the felt, there to be absorbed. This combination proved ineffective because the reflection from such surfaces was too great, and consequently very little sound was absorbed.

There are other factors than the absorption coefficient to be weighed in the selection of an absorbing material. Among these are the space available, the cost, sanitation, durability and fire and water resistance. Sanitation should receive special consideration. Porous materials collect dirt readily and are difficult to clean, and unfortunately it is a characteristic of a good absorbing material that it be porous. The removal, by means of filters, of as large a portion as possible of the foreign matter from the air before it comes in contact with the sound absorbing material, is a partial solution. In considering the space available for an absorbing material, it has been found at the Riverbank Laboratories, Geneva, Illinois, that long narrow strips of an absorbing material are more effective per unit of exposed area than broad panels of the same material.

Selection of Materials

There is a wide variety of absorbing materials from which to choose, and some of these will not greatly impede the air flow. Perforated metal with 1/16 in. holes, 64 per square inch, when backed up by an absorbing material such as rock wool, 1 in. thick, has a coefficient of 0.84 at a frequency of 1,024, based on tests conducted by the U. S. Bureau of Standards. Laboratory tests have shown this to be a particularly good combination with little loss in volume of air delivered.

The absorption of sound has been measured by several methods and the absorption coefficients calculated for a great variety of materials. The absorption coefficients of materials tested by the U. S. Bureau of Standards and other laboratories are useful in the choice of an absorbing material.

The most effective thickness of absorbing material depends on the wave frequency of the sound. The absorption coefficients increase with the increase in thickness of material up to a point of maximum effectiveness which is attained at smaller thicknesses for the greater frequencies. For instance, in the higher ranges of 2,000 or over, almost no more absorbing power is obtained by increasing the thickness of felt over ⅜ in. In relation to the cost and to the space available, the law of diminishing returns becomes operative at some point less than the point of maximum absorbing power.

Mufflers, such as are used on automobiles, seem to have no practical application in unit ventilators. This type of silencer must necessarily occupy considerable space and must present to the air a prohibitive degree of air resistance.

The arrangement and the design of baffles and dampers within the cabinets of unit ventilators has considerable effect on the noise output of these machines because these surfaces are apt to reflect and to focus the sound in an undesirable manner. If the machine discharge is directed vertically the noise is less noticeable than if it is directed toward the occupants of the room. Refraction of the sound, due to the difference in velocities and temperatures of the air currents in the room, changes, although to a minor degree, the direction in which the sound travels.

In the design of ventilating equipment it is often possible to fill in sharp corners and other turbulence-producing places with an absorbing material which will not only be effective in reducing noise but may actually reduce the total resistance to the flow of air. Recently a number of experimental machines were lined with one of the commercial absorbing materials. The material is composed of cedar excelsior which is hydraulically pressed into slabs while immersed in a magnesite cement. After baking, an extremely effective porous absorbing material is produced, which is permanent, fire proof and easy to apply.

The excelsior-magnesite material was placed in the machines in such a manner that it filled in several sharp corners. As a result, the air delivery of the machines was 2 per cent greater than before, and an appreciable reduction in the noise output was observed.

Sometimes canvas connections are placed between unit ventilators and duct work. This canvas yoke assists in reducing noise in two ways; it prevents the transmission of mechanical vibration between the machine and the duct work, and, also acts as an absorber.

The transmission of sound through the fan housings and the cabinets of unit ventilators is largely governed by the inertia and rigidity of these barriers. The amplitude of a sound vibration is very small, usually less than 0.005 in., depending on the loudness of the sound. If the walls have sufficient rigidity and inertia most of this movement will be prevented, and the transmission of sound through the walls thereby prevented.

Sound Interference

The phenomena of sound interference is of particular importance in unit ventilators. In most unit ventilators there are two fans assembled upon the shaft of a motor between them. These fans constitute two continuous sources of sound of the same amplitude and pitch, that is, the same frequency of alternation, pressure and rarefaction. Under these conditions at any instance there will be locations where the pressure from one source will exactly neutralize the partial vacuum from the second source, and therefore no sound will be audible. At one-half of a vibration later, these conditions will be reversed but the waves will still neutralize each other. At other and different points the two condensations may arrive simultaneously, at which point particularly loud sounds will be heard.

Due to the fact that there are several sources of sound in a unit ventilator and also that these sounds are of different pitch, the effect of interference is only partial, that is, there will never be complete neutralization at any point in a room of installation of one of these units, but there will be localities where the sound intensities will be changed by this interference and made louder or softer. Interference is also to be expected from the reflection of the sound waves both in the unit itself and from the walls of the room of installation. The interference between the two sound sources also has a reaction upon the sound output of those sources. This effect has been taken advantage of in the design of sound filters which prevent the emission of sound at those frequencies for which there is attenuation. No application has been made of such sound filters to unit ventilators so far as the author knows.

Resonance plays a part in the internal design of unit ventilators. Certain dampers and baffles are likely to be set in vibration by the sound waves and will then reinforce the frequency with which it is in tune.

The effect of reverberation inside the unit cabinets is practically negligible. It is proportional to the volume of the cabinet, and this is relatively small. The effect of reverberation in the room of installation, may, however, be great, although that is a matter which cannot be aided by any changes in the ventilating unit design.

Usually ventilating units are tested, and sometimes judged for noise, in the empty bare room of installation. This is not a fair test as it is not under these conditions that the machine will ordinarily be operated. The presence of an audience or of a class of students, with their large areas of sound absorbing clothing, will make an appreciable difference in the results of such a test.

Conclusions

The reduction of noise is a major problem in unit ventilating. If it were not for the restraining influence of noise, it would be possible to manufacture much more compact machines for delivering the same volume of air. Noise, therefore, has a direct and limiting effect on the entire design of these machines. Progress in noise reduction means progress in all phases of unit ventilator development.

National Power Show

Under the leadership of I. E. Moultrop, Chairman, preliminary details of the Ninth National Exposition of Power and Mechanical Engineering to be held December 1 to 6, 1930, in Grand Central Palace, New York City, are well underway. It has been stated by the committee in charge that they are already confronted with the problem of finding space for new exhibitors.

Business conditions in general are such as to make a show at this time of more than usual value. In the first place, a larger attendance than ever is expected because the average executive feels the need for keeping up to the minute in all methods of economy in these times and on the other hand business is not rushing at a pace to keep the important persons in the larger organizations too busy to attend.

That power is the source of all wealth in the final analysis is a fact that the manufacturer who is turning it into the products that are used daily realizes and he is realizing it more and more intelligently each year, as the economies that have been effected by those who have equipment to sell have been translated into dividends for such manufacturing concerns as have been keen enough to adopt new devices and improved methods.

When the first Power Show opened its doors nine years ago for instance, the use of pulverized coal was in its infancy and the field was divided in opinion relative to its use. Yet today it is stated that in the electric and public utility plants of this country alone there is being burned each year 7,112,739 tons of coal in pulverized form in devices that were looked upon as a curiosity in 1921.

The forty-two million tons of coal each year burned in the power plants of the public utilities in this country must have some approved mechanical means of being turned into power and all of the stoking devices will be shown at the show. The underfeed stokers that burn over 20,000,000 tons a year will be shown with chain grates that are responsible for the consumption of 12,000,000 tons. In this connection it is interesting to note that of the 42,000,000 tons of coal burned in the public utility field, only 973,000 tons are burned by hand-fired methods or open grates.

The engineering profession is moving ahead so fast in America that it is never possible to forecast the new devices that will appear within the span of a year. Many manufacturers have taken advantage of the lull of the past months to develop new devices and improvements on old ones that will be shown for the first time at the Power Show this year.

The character of the Advisory Committee indicates the interest and ramifications of the exhibits and their importance to the several engineering professions and industries. They are Homer Addams, Past President of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, N. A. Carle, Manager the Pacific Electric and Mfg. Co., H. D. Edwards, President, *American Society of Refrigerating Engineers,* Fred Felderman, Past National President, *National Association Stationary Engineers,* Vincent M. Frost, Chairman Power Division, *American Society of Mechanical Engineers,* L. A. Harding, President, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, C. F. Hirshfeld, Chief Research Department, Detroit Edison Company, O. P. Hood, Chief Mechanical Engineer, U. S. Bureau of Mines, John H. Lawrence, President Thomas E. Murray Company, Fred R. Low, Past President, *American Society of Mechanical Engineers,* David Moffatt Myers, Consulting Engineer, Fred W. Payne, Co-Manager Exposition, Charles Piez, President, *American Society of Mechanical Engineers,* Joseph W. Roe, Chairman Professional Division, *American Society of Mechanical Engineers,* Charles F. Roth, Co-Manager of the Exposition, and President of the *National Electric Light Association.* These men will again be sponsors of the Power Show under the able leadership of I. E. Moultrop as Chairman.

It is interesting to note that every state in the union was represented on the visitors list of those signing at the door last year. The growth in interest in this field is apparent when it is remembered that there were only 105 exhibitors at the first show while according to present indications, there will be more than four hundred exhibitors at the 1930 show. Since the National Power Show will not be held in 1931, it is expected that the forthcoming Exposition will draw better exhibits and a record attendance of visitors.

NOMINATIONS FOR OFFICERS OF THE SOCIETY FOR 1931

The Nominating Committee appointed to select candidates for Officers of the Society for the coming year, 1931, takes pleasure in submitting the following list of nominees:

For President:
W. H. Carrier, Newark, N. J.
For First Vice-President:
F. B. Rowley, Minneapolis, Minn.
For Second Vice-President:
W. T. Jones, Boston, Mass.
For Treasurer:
F. D. Mensing, Philadelphia, Pa.

For Members of the Council:

Three-Year Term

D. S. Boyden,* Boston, Mass.
E. K. Campbell, Kansas City, Mo.
E. O. Eastwood, Seattle, Wash.
Roswell Farnham, Buffalo, N. Y.
E. Holt Gurney, Toronto, Ont.

Respectfully submitted,

NOMINATING COMMITTEE

G. L. Larson, *Chairman*

Chapters	*Representative*
Cleveland	H. M. Nobis
Illinois	J. F. Hale
Kansas City	N. W. Downes
Massachusetts	J. F. Tuttle
Michigan	J. H. Walker
Minnesota	G. C. Morgan
New York	H. B. Hedges
Western New York	O. K. Dyer
Ontario	A. S. Leitch
Pacific Northwest	E. O. Eastwood
Philadelphia	R. C. Bolsinger
Pittsburgh	T. M. Dugan
St. Louis	C. A. Pickett
Wisconsin	G. L. Larson

In accordance with the provisions of the Amended Constitution, as given, ballots containing the names of the above candidates will soon be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. 9—Section 2. The Nominating Committee shall consist of a member designated by each Chapter, which shall effect its own organization and elect its own Chairman.

Section 3. The Committee may meet at the call of the Chairman, for discussion, but the official meeting shall be held at the Semi-Annual Meeting of the Society, at which time the candidates for each of the offices of the Society to be filled at the next Annual Meeting shall be selected by a majority vote of the members of the Committee present and voting. Five shall constitute a quorum.

Section 4. The Committee shall first secure the consent of all candidates selected, and shall, if possible, announce the names of the candidates at the Semi-Annual Meeting, but in any case the names of the candidates shall be certified to the Secretary of the Society at least four months before the next Annual Meeting. The Secretary shall publish these names in the October issue of The Journal.

Art. 10.—*Section* 3. The Secretary shall prepare ballots with the names of all candidates and forward them to the members at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election*. The election is governed by the By-Laws for the election of officers, with the single exception, that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee.

In accordance with the Regulations for the government of the Research Laboratory, adopted at the January, 1919, Meeting, and revised January, 1930, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January, 1931:

C. A. Booth, Buffalo, N. Y.
F. B. Howell, New York, N. Y.
Walter Klie, Cleveland, O.
F. B. Rowley, Minneapolis, Minn.
J. H. Walker, Detroit, Mich.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

SECTION I—ORGANIZATION

Research Committee

1. There shall be a standing committee known as the Research Committtee, consisting of fifteen members, each serving for three years, and five retiring each year.

a. The Council shall nominate previous to October first of each year five members to fill the vacancies of those retiring at the next Annual Meeting.

b. The nominations made by the Council shall be published in the October issue of the Society's Journal.

c. Prior to December first of any year, any ten members, over their own signatures, may nominate one or more additional members of the Research Committee, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

d. The election shall otherwise conform to the regulations provided for the election of officers of the Society.

e. Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.

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*Nominated to fill an unexpired term for two more years beginning 1931.

IN COMPILING our own "Who's Who," among the candidates for Council membership in 1931, some facts, both serious and otherwise, have been obtained from reliable sources and are passed along to the Society members in the form of biography, autobiography and sometimes a mixture of both.

D. S. Boyden

Davis Stuart Boyden was born December 28, 1872, at Walpole, Mass., and attended the public schools for his preliminary education. After leaving school, Mr. Boyden specialized in the field of light, heat and power and his entire business career dating from 1890 has been devoted to this work. For two years he was with the Suburban Light, Heat and Power Co. in Boston, and then for fifteen years was engineer and superintendent at the Block Plant Electric Light Co., where he laid out and supervised the extensive heating business carried on by this company.

D. S. Boyden
Boston, Mass.

Mr. Boyden has acted as a consultant on numerous district heating plants and since 1907 has been superintendent of steam heating department of Edison Electric Illuminating Co. of Boston, where he supervised the estimating, construction and operation of the company's extensive Steam Heating Service Plants.

For diversion, the Colonel (yes! he is a Lieutenant Colonel in the O. R. C., U. S. Army), plays golf and plays with radio telegraphy.

Mr. Boyden has been a member of the Society since 1909 and first served on the Council in 1917. He has served this year to fill an unexpired term and now has been nominated to serve two more years beginning 1931.

E. K. Campbell

Mr. Campbell was born in Newton, Iowa, December 30, 1874, and he says his birthday has come so close to Christmas that he has never received any birthday presents.

After graduating from grade and normal schools, he received the degree of Bachelor of Science from Grinnell College in 1900.

From boyhood up he worked in the furnace business with his father, who contributed the idea of recirculation and of air moistening to the furnace business. From 1900 until 1909 he conducted a branch office for his father's company in Kansas City, leaving that company and establishing his own business January 1, 1910, with the avowed purpose of developing a fan furnace system on the basis of recirculation particularly for churches and other buildings having large rooms. He experienced all the trouble incident to developing something new, but stayed with his troubles and won out.

E. K. Campbell
Kansas City, Mo.

Mr. Campbell's only technical association is with the American Society of Heating and Ventilating Engineers, but he is a member of the local Engineers Club, has been a director of the City Club, is now a director of the Citizens League and at the present time is president of the Kansas City Chapter of the A.S.H.V.E., and as such is a member of the President's Round Table of Kansas City.

E. O. Eastwood

Everett Owen Eastwood was born in Portsmouth, Va., February 5, 1876, and after graduating from the University of his native state in 1896 with a degree of Civil Engineer, he returned to the university and took degrees of Bachelor of Science in 1897 and Bachelor of Arts and Master of Arts in 1899.

In 1902 Professor Eastwood received the degree of B.S. in Naval Architecture from the Massachusetts Institute of Technology and for the next few years devoted his time to shipping interests with the scientific department of the Bureau of Construction and Repairs, U. S. Navy, Washington, D. C., and with the Fore River Shipbuilding Co. In 1904 Professor Eastwood began his teaching career at the Lehigh University as Instructor of Naval Architecture and Mechanical Engineering and since 1905 has been a member of the faculty of the University of Washington, where he has served as Professor of Mechanical Engineering and head of the department for the past quarter century. He is also Director of the Department of Aeronautical Engineering. In addition to his university work, Professor Eastwood is consulting engineer for the university and has been connected with various business enterprises, fraternal and engineering organizations.

E. O. Eastwood
Seattle, Wash.

At the present time Professor Eastwood is President of the Pacific Northwest Chapter of the A.S.H.V.E. and was instrumental in its organization several years ago. He is a member of the A. S. M. E., a Charter Member of the Washington section and in addition to serving the local section in several capacities, he has served the national organization on the Board of Managers, 1923 to 1926, and was vice-president from 1926 to 1928. He is a member of Sigma Xi, Tau Beta Pi and Sigma Alpha Epsilon.

Professor Eastwood has been a member of the Society since 1921 and has been nominated to serve a three-year term on the Council commencing 1931.

Roswell Farnham
Buffalo, N. Y.

Roswell Farnham

Roswell Farnham was born August 13, 1890 in Buffalo, N. Y., and after graduating from the public schools of the city, spent four years in the Granite State at the University of Vermont, where he received the degree of B.S. in Mechanical Engineering in 1913.

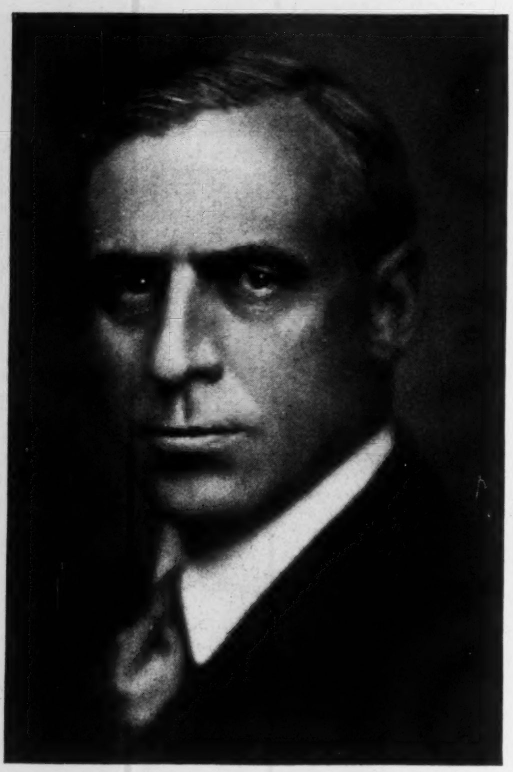
Immediately upon graduation from college, he started with the Buffalo Forge Co. and is now District Manager of Engineering Sales with headquarters in Buffalo.

It might be accurately said that Ros, as he is familiarly known, has grown up in Buffalo and except for his brief absence to attend college and the less extended ones to attend A.S.H.V.E. meetings, he has spent all of his business career in his native city, where he is active in local engineering and fraternal affairs.

Soon after Ros joined the Society in 1920 he was put to work and served for five years as secretary of the Western New York Chapter and was elected president in 1926, when the Society held its last annual meeting in Buffalo. When the Engineering Society in Buffalo needed a capable secretary, Ros was drafted to serve from 1925 to 1927 and this year he is serving as president of the Engineering Society of Buffalo.

When given the Edison Questionnaire, Ros admitted that he had missed attending only one Chapter and Society meeting since 1920; that Buffalo would soon win the Little World Series, and that golf was a great game, even though you were no Bobby Jones.

Mr. Farnham has served one term on the Council and has been renominated for the three-year period beginning 1931.



E. H. Gurney
Toronto, Ont.

E. H. Gurney

Edward Holt Gurney was born in Toronto, Canada, August 25, 1882, and received his preliminary education at the Model School and Jarvis Street Collegiate, after which he attended the University of Toronto.

His first work in the heating and ventilating field was with the Gurney Foundry Co., Ltd., and in 1919 he became president of the concern, except for a two-year period during the War, when he was a director of the Imperial Munitions Board. He is thoroughly familiar with the heating industry, both in Canada and the United States. He was president of the Gurney Heater Mfg. Co. in Boston from 1919 to 1927. In addition to his activity as a member of the Toronto Chapter of the Society he has served as Director of the Ontario Research Foundation, the Canadian National Exhibition and the Toronto General Hospital.

Mr. Gurney has been nominated to serve for three years on the Council beginning 1931.

Harry Morton Dix Dies

Harry Morton Dix, well-known heating and ventilating engineer in New England, died at his home, 11 Elmwood St., Worcester, Mass., on September 14, 1930.

Mr. Dix was born on November 11, 1883, in Charlestown, Mass., where he attended the elementary and high school and the International Correspondence School.

In 1905 he became associated with the B. F. Sturtevant Co., Hyde Park, Mass., where he remained until 1909. After being identified for about six months with the Massachusetts Fan Co., at Watertown, he resumed his connection with the B. F. Sturtevant Co. for another year, when he journeyed to Montreal as an engineering and sales representative for that company. He was also connected with the following concerns during his engineering career: Eadie-Douglas, Ltd., the Canadian Domestic Engrg. Co., the Central Supply Co., and at the time of his death was engineer for the American Radiator Co., Providence, R. I.

~~Mr.~~ Dix joined the Society in 1925 and became active in Society affairs in New England, where he will be missed greatly by his host of friends among the architects and engineers.

The officers and Council of the Society express their sincere sympathy to his widow, Grace Patten Dix, and his son, Harry M., Jr., who survive.

A. G. A. Offers Samuel Insull Award for Outstanding Progress

The *American Gas Association* has announced the offering of a special annual award to the member gas company of that organization, which has made, during the previous calendar year, the most outstanding progress in distinguished public service. This award has been made possible by Samuel Insull, of Chicago, Illinois, and will be known as the Samuel Insull Award for Distinguished Public Service.

The first Samuel Insull Award will be for the calendar year 1931 and will be presented at the 1932 Annual Convention of the *American Gas Association.* All member companies of the Association in the United States, its possessions or in Canada, whether large or small, whether distributing manufactured, natural or mixed gas, will be invited to submit exhibits for consideration of a special committee which will have charge of examining the exhibits and selecting the winning entrant annually.

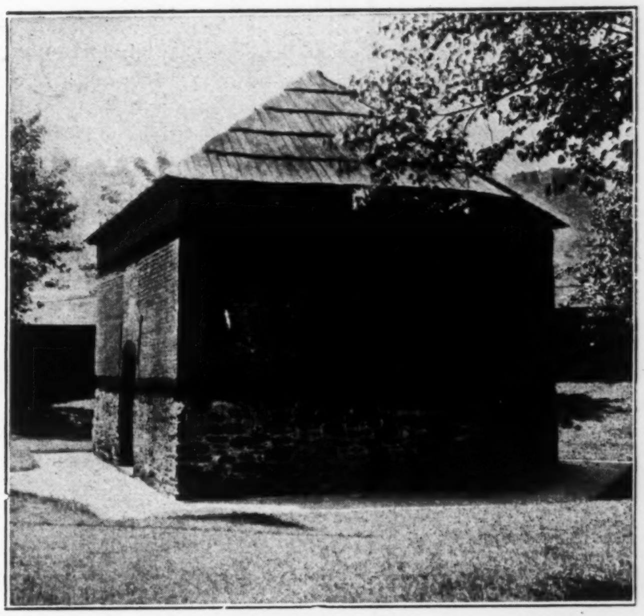
In order to make the comparisons of company accomplishments more valid and significant, the following classifications have been established for companies submitting exhibits:

Class A—Companies having 125,000 or more meters.
Class B—Companies having more than 40,000 and less than 125,000 meters.
Class C—Companies having less than 40,000 meters.

The award for 1931 will be available to companies in *Class A*—those having 125,000 or more meters. The award for 1932 will be available to companies coming within *Class B,* and that for 1933 will be available to companies within *Class C.* In subsequent years the award will be rotated in the order indicated.

The winning company will receive an engrossed certificate suitable for mounting in the company's offices as public evidence of its company achievements, and in addition will be entitled to designate three members of its personnel who may attend one of the special courses for business executives such as the course in Public Utility Management and Economics offered during the summer months by the Graduate School of Business Administration, Harvard University, or other courses offered by similar institutions. Those individuals designated by the winning company will be entitled to free tuition and expenses during the period of the course at the institution selected.

Pittsburgh Plans for 37th Annual Meeting



Block House at Fort Pitt
Built 1759

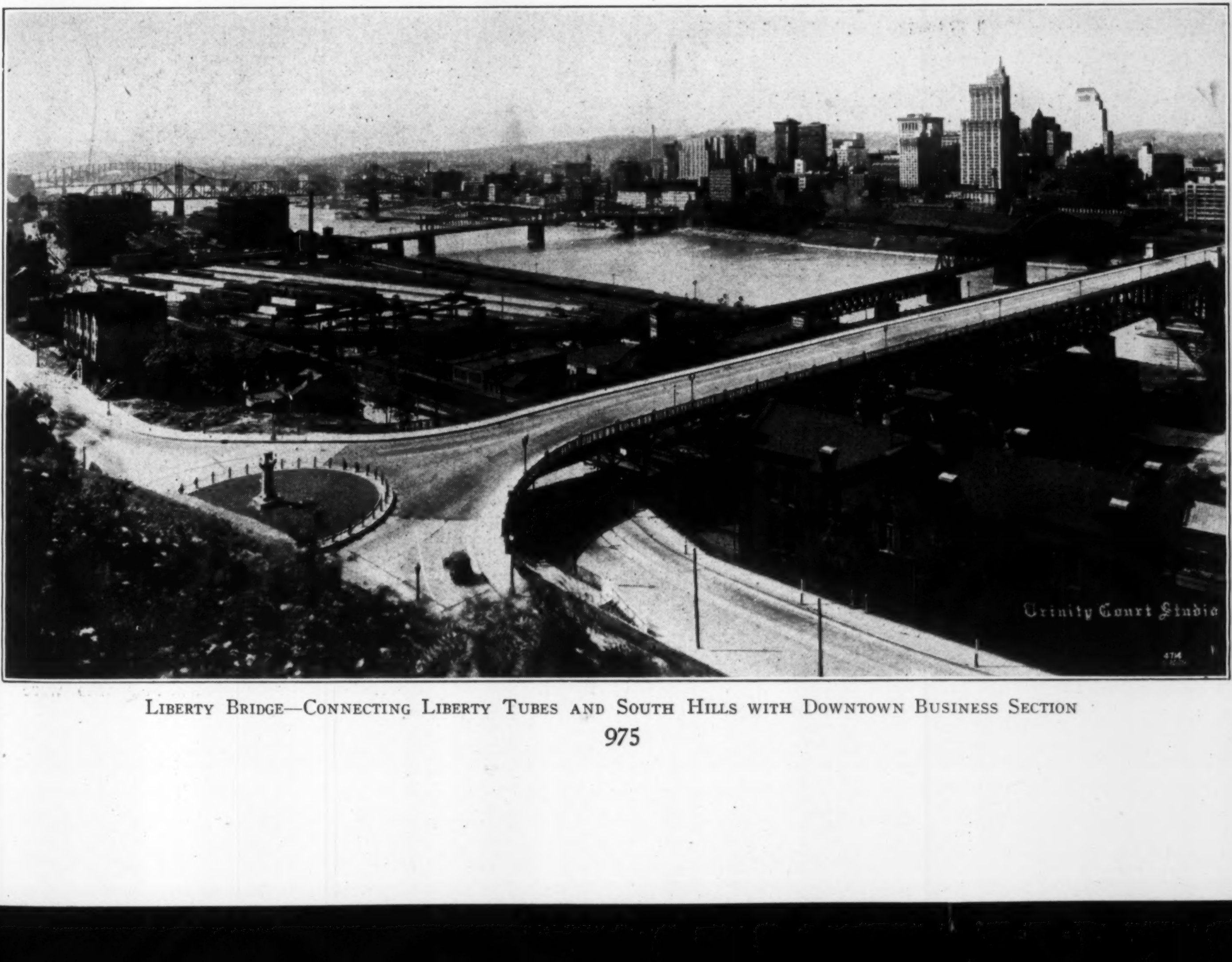
PITTSBURGH is planning a warm welcome for the members of the Society, who will gather for the 37th Annual Meeting at the William Penn Hotel, January 26 to 29, 1931. The Pittsburgh Chapter members will be hosts and arrangements are in the hands of a general committee working under the leadership of H. Lee Moore. Each of his assistants on the general committee will head a special committee as follows:

H. B. Orr, Chairman Reception Committee.
F. C. Houghten, Chairman Program Committee.
R. B. Stanger, Chairman Entertainment Committee.
F. A. Gunther, Chairman Banquet Committee.
W. W. Stevenson, Chairman Transportation Committee.
J. E. McGinness, Chairman Publicity Committee.
F. C. McIntosh, Chairman Finance Committee.

Pittsburgh is so centrally located that it is only a night's journey from the majority of chapter cities and a large attendance is anticipated. Pittsburgh is generally thought of as the Steel City, as it has been the leading producer of iron and steel from the earliest days.

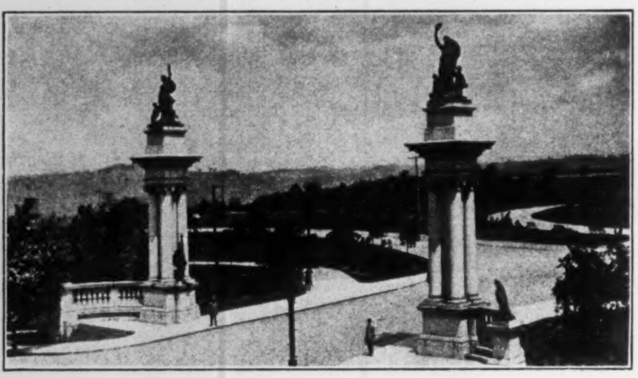
The business section of the city is located at the junction of the Allegheny and Monongahela Rivers and the meeting headquarters at the William Penn Hotel are conveniently located to rail terminals and the main automobile routes, the Lincoln and William Penn Highways. Modern air transport service offers fast schedules from many large cities of the middle west.

Pittsburgh is a great industrial laboratory, so that it is fitting to have the Society's Research Laboratory located



Liberty Bridge—Connecting Liberty Tubes and South Hills with Downtown Business Section at the Pittsburgh Experiment Station of the U. S. Bureau of Mines. It is planned to spend one day at the Laboratory, so that members may see the laboratory equipment in operation after attending a technical session in the morning.

On Monday, January 26, President Harding will call the opening session to order at 2:00 p.m., in the William Penn Hotel. During the morning of that day Council and Committee Meetings will be held and registration will be started.



ENTRANCE TO HIGHLAND PARK

On Tuesday, January 27, two technical sessions, one in the morning and one in the afternoon, will be held at the William Penn Hotel and Wednesday will be spent at the Laboratory in the U. S. Bureau of Mines, where a technical session will convene in the morning and inspection of the experiment station and the Society's Laboratories is planned for the afternoon.

On Thursday, January 29, two technical sessions will be held at the William Penn Hotel.

The Program Committee has selected a fine group of papers for presentation and they will be printed in the JOURNAL in advance of the meeting.

Among the subjects proposed for discussion are the following:

- Capacity of Return Mains for Steam and Vapor Heating Systems, by F. C. Houghten and Carl Gutberlet.
- Welded Piping for Building Heating Systems, by F. G. Outcalt.
- Domestic Heating by Electricity, by Elliott Harrington.
- Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces, by L. W. Schad.
- Utilization of Waste Heat, by Perry West.
- Noise Reduction in Ventilating Units, by Warren Ewald.
- Some Studies on the Prevention of Noise in Ventilating Ducts, by G. L. Larson.
- Developments in Heating and Ventilation During 1930, by S. R. Lewis.

Something different in the way of an entertainment program is promised by the Pittsburgh Chapter Committee and the big social event of the meeting will be the Annual Banquet at the William Penn Hotel, on Wednesday night, January 28. An interesting program for the ladies is in prospect and those who attend will find much to interest them in Pittsburgh, which in addition to its great manufacturing industries, has such famous educational institutions as Carnegie Institute of Technology, the University of Pittsburgh, Mellon Institute and several art galleries, among which is the world-renowned Carnegie Institute fostered by the great iron maker. One of Pittsburgh's historic spots is the Block House built in 1764 at Ft. Pitt, formerly Ft. Duquesne.

All eyes will be focused on Pittsburgh for the four days, January 26 to 29, 1931 and the Pittsburgh Chapter sends an urgent invitation to plan for the Annual Meeting.

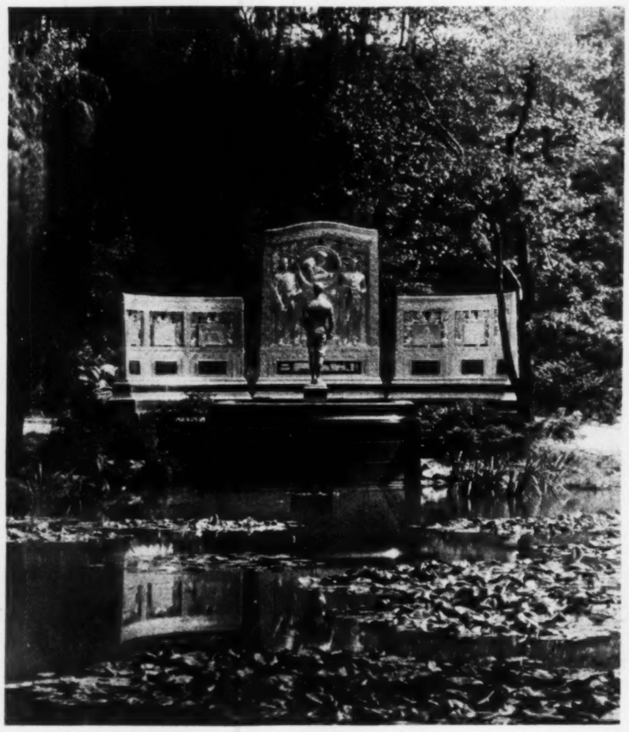
Westinghouse Memorial Dedicated

Leaders of industry, business, and scientific research paid homage to the late George Westinghouse, founder of the many Westinghouse industries in the dedication of a memorial in Schenley Park, Pittsburgh, Pa., on Monday, October 6, 1930.

The dedication ceremony was presided over by E. M. Herr, vice chairman of the Board of Directors of the Westinghouse Electric & Mfg. Co. The program included addresses by A. L. Humphrey, president of the Westinghouse Air Brake Co. and one of the leaders in the movement for providing the memorial; James Francis Burke, former congressman, and Bishop Alexander Mann of the Pittsburgh Episcopal Diocese.

Following these addresses, George Munro, representing the Westinghouse Veterans, presented the memorial. After the monument was unveiled, Mayor Charles H. Kline made the speech of acceptance in behalf of the City of Pittsburgh.

The central panel of the memorial shows the figure of George Westinghouse leaning over his drawing board between the figures of the skilled mechanic and the engineer. The center piece is a ten-foot figure representative of American youth. The right-hand panel depicts three of Mr. Westinghouse's most notable achievements, the illumination of the Chicago's World's Fair, the air brake, and the modern railway signaling system. The panel at the left of the central group shows graphically three additional accomplishments of the great inventor, namely, the alternating-current railroad electrification, the steam turbine, introduced into America by Mr. Westinghouse, and the hydroelectric plant at Niagara Falls. The memorial is unique in that the backs as well as the fronts of the workman and the engineer are shown. These become full sculptural figures set in an openwork of panel formed by a delicate lacework of oak leaves. The *full relief* idea was carried out in the subsidiary panels so as to make a view of the memorial from the rear as interesting as from the front.



WESTINGHOUSE MEMORIAL, SCHENLEY PARK, PITTSBURGH, PA.

This memorial represents the combined efforts of Henry Hornbostel, architect, Daniel Chester French, sculptor, and Paul Fjelde, sculptor, as well as other talented artists who were associated with them.

Local Chapter Reports

Cleveland

The December meeting was the first of the season held by the Cleveland Chapter during 1929-30. The speaker of the evening was H. M. Nobis, who discussed the heating and ventilating system in the Allen School at Elyria, Ohio, which the members and their guests enjoyed.

This was followed by the meeting held on February 19, 1930, in the Cleveland building of the Johnson Service Co. M. F. Rather delivered an interesting address on Temperature and Humidity Control, which was followed by a bridge contest and a buffet supper. This was one of the most interesting meetings of the year for the Cleveland Chapter.

The next meeting was held on April 17, 1930, with Prof. A. P. Kratz as the guest speaker of the evening. Professor Kratz selected as his topic, Progress of Research, which he rendered in a very interesting manner.

At the May 26, 1930, meeting the chapter was greatly honored by having Pres. L. A. Harding and F. C. Houghten, Director of the Society's Research Laboratory, among their guests. President Harding spoke on the Utilization of the Sun's Energy, while Director Houghten discussed the Heat Absorption from Solar Radiation.

The officers serving the Chapter during 1929-30 are as follows:

President, F. H. Morris.

Vice-President, A. R. Bruggeman.

Secretary, R. G. Davis.

Treasurer, H. M. Nobis.

Board of Governors, C. F. Eveleth, T. A. Weager and W. C. Kammerer.

Illinois

The Illinois Chapter of the Society has completed a very successful year under the guidance of the following officers:

President, H. G. Thomas.

Vice-President, T. H. Monaghan.

Secretary, C. W. DeLand.

Treasurer, C. W. Johnson.

Board of Governors, J. H. O'Brien, M. S. Good and E. P. Heckel.

The *Nominating Committee* consisted of the following members: W. A. Mertz, L. L. Narowetz, Jr., and Geo. Blanding; and the *Board of Tellers,* B. L. Casey, W. D. Graham and John F. Hale.

The future of the American Society of Heating and Ventilating Engineers was ably presented at the October 1929 meeting by Thornton Lewis, at that time President of the Society.

Welding was the topic of discussion at the November meeting, the speaker of the evening being F. G. Outcalt, resident engineer in the Development Section of the Linde Air Products Co., representing the *International Acetylene Association.* Mr. Outcalt gave a very excellent talk and presented slides to illustrate his talk.

The December gathering of the Illinois Chapter members and their guests was one of great interest. A series of talks was given on the recent development in thermostatic control by representatives of the following concerns: American Radiator Co., Minneapolis-Honeywell Co., National Regulator Co., F. I. Raymond Co., Johnson Service Co., Powers Regulator Co., and the Conco Temperature Control.

E. P. Heckel, the speaker of the January meeting of the Illinois Chapter, described the ventilating, air conditioning and refrigerating equipment at the Chicago Stadium.

The February meeting of the Chapter was held at the Hotel Sherman, with an attendance of 109 members and guests, and proved to be one of great interest. H. R. Linn rendered a very informative talk on the status of oil burners in the industry.

Prof. A. C. Willard, of the University of Illinois, and Prof. A. P. Kratz were the guest speakers of the evening at the March meeting, which was devoted to Room Heating, or What is Wrong with 70 F Breathing Line Temperature.

The Society and Research was the topic of discussion at the April meeting, which was presented by Prof. F. B. Rowley of the University of Minnesota. This was the last meeting of the 1929-1930 season for the Illinois Chapter, the meetings being dispensed with until October 13, 1930.

Kansas City

September 8, 1930. The first fall meeting of the Kansas City Chapter of the Society was held at the Ambassador Hotel, and was attended by 30 members and guests. After dinner Pres. E. K. Campbell called the meeting to order and called for volunteers from the Kansas City Chapter to prepare papers to be presented before the Society at its Annual Meeting in January at Pittsburgh.

Following the regular order of business C. A. Burton gave an enlightening talk on his experience as a member of the Water Supply Committee in conjunction with the ten year plan, which was one of the most interesting talks of the evening.

J. A. Kitchen then recommended that a clause be inserted in the By-Laws of the Chapter to provide for an acting Chapter Secretary in the event of the absence of the duly elected Secretary. He also suggested eliminating Section 5, and further suggested that the By-Laws be rewritten to include these changes and be submitted for acceptance and adoption at the November meeting.

The report of the Summer Meeting of the Society was then presented by N. W. Downes, and this was followed by an interesting discussion on the Willard and Kratz paper by E. K. Campbell, Jr.

Mr. Lynch, Director of the Safety Council and guest of the Chapter, was introduced by J. M. Arthur, Jr., and with the McClintock report on Traffic as his theme, held the attention of all present until adjournment at 10:15 p. m.

Minnesota

The Minnesota Chapter of the Society has enjoyed a number of interesting meetings during the year, which began with the October 1929 meeting.

At the October meeting, Thornton Lewis, at that time the President of the Society, delivered an interesting address on the future of the American Society of Heating Ventilating Engineers. C. W. Farrar, chairman of the Membership campaign, covered plans for increasing the membership of the Society. A. V. Hutchinson, Secretary of the Society, then outlined the program of the Annual Meeting in Philadelphia.

The next meeting was held in November and was in the form of a social event which took place at the Oak Grove Country Club.

The December meeting consisted of an inspection trip through the plant of the Flax-li-num Insulating Co. Prof. F. B. Rowley, University of Minnesota, gave an interesting review on Research on Heat Transmission and problems confronting the manufacturers of insulation. R. L. Rundorf, heating engineer of Northern States Power Co., gave a paper on Gas Heating and Insulation, which was followed by a discussion on the Past, Present and Future of Insulation, given by Frank Evans.

Homer R. Linn presented an interesting talk on the Status of the Oil Burner at the January meeting.

At the February meeting Albert Buenger reported on the Annual Meeting held at Philadelphia, and this was followed by a discussion on the proposed boiler code. Webster Tallmadge, New York, then spoke on Zone Control of Heating.

A. B. Dixon was the guest speaker at the March meeting, and selected as his topic, Unit Heaters and Copper Radiation.

The April meeting was also in the form of an inspection trip. St. Mary's Hospital and the Franklin Heating Station at Rochester were visited. Neil Adams gave a brief talk on the Franklin Heating Station, and later Mr. Hutchinson, Secretary of the Society, outlined the duties of the Chapter at the Semi-Annual Meeting in Minneapolis.

The May meeting consisted of the regular annual golf tournament and the election of officers for the season 1930-31.

The officers elected to serve the Chapter are as follows:

President, D. M. Forfar.
Vice-President, H. E. Gerrish.
Secretary-Treasurer, W. F. Uhl.
Board of Governors, E. F. Jones and M. S. Wunderlich.

The Committees are as follows: *Meetings Committee:* M. H. Bjerken, *chairman*; G. A. Dahlstrom, Carrol Lewis, Fred Shernbeck, W. N. Parks, W. B. Clarkson and A. M. Wagner; *Membership Committee:* Albert Buenger, *chairman;* E. F. Jones, *captain;* A. J. Huch, *captain; Publications Committee:* M. S. Wunderlich, *chairman;* Charles Foster and F. B. Rowley; *Nominating Committee:* E. J. Jones, *chairman;* A. J. Huch and Albert Buenger; *Auditing Committee:* H. H. Bradford, *chairman;* J. M. Heagler and C. E. Gausman; *Legislative Committee:* S. A. Challman, *chairman;* H. M. Betts and A. B. Algren.

Massachusetts

September 8, 1930. The Board of Governors held its first fall meeting at the Boston City Club on September 8, 1930, with Messrs. Clough, Gifford, Jones, Kellogg, McCoy, Moulton and Webb in attendance.

President McCoy called the meeting to order and after a short discussion it was voted that D. S. Boyden be appointed Chairman of Arrangements for the Semi-Annual Meeting of the Society to be held at the New Ocean House, Swampscott, during the last week in June, 1931. Mr. Boyden's appointment is to carry with it the power to select his own committee.

The resignation of Mr. Webb as Secretary and member of the Board of Governors was then presented and accepted.

Leslie Clough was then nominated and elected to fill the vacancy as Secretary, which was followed by adjournment.

Western New York

At the first meeting of the year 1930, which was held in January, the election of the following officers took place:

President, F. H. Burke.
First Vice-President—Joseph Davis.
Second Vice-President, M. C. Beman.
Treasurer, C. H. Love.
Secretary, D. J. Mahoney.

The *Board of Governors* is composed of: R. T. Coe, L. A. Harding, C. W. Farrar, Hugo Hutzel, W. G. Fraser, Roswell Farnham, M. S. Jackson, C. A. Evans and O. K. Dyer.

The *Speakers Committee* is as follows: O. K. Dyer, *chairman;* L. A. Cundall and A. K. Tinker.

The *Entertainment Committee* consists of the following: J. J. Landers, *chairman;* L. N. Castin and H. B. Roarke.

The following comprises the *Membership Committee:* Joseph Davis, chairman; W. F. Johnson and Wm. Roebuck, Jr.

The February meeting was a dinner-dance at the Hotel Buffalo, which was an evening of enjoyment for the members and guests who attended.

Lewis J. Summerhays, Rochester, spoke on the Design and Construction of Chimneys at the March meeting.

The April meeting was devoted to the subject of the sun as the source of all energy, which was ably presented by Prof. J. H. Curtin of D'Youville College, Buffalo, one of Buffalo's prominent astronomers.

The May gathering was termed "Kentucky Night." Dean Anderson of the University of Kentucky, Lexington, Ky., with his senior engineering class were the guests of the evening.

The annual mid-summer dinner dance was held in place of the June meeting of the Western New York Chapter, which took place at the Automobile Club, Clarence, N. Y.

All the meetings of the year have been marked by a splendid attendance.

Ontario

A. S. L. Barnes, engineer for the Hydro-Electric Power Commission of Ontario, was the first speaker of the season. Mr. Barnes addressed the members and guests at the January meeting on the Possibilities and the Extent to which Electric Heating Can be Carried.

At the February meeting, Professor Angus of the University of Toronto spoke on the Cooperation of the University of Toronto and the Society in Solving Mutual Problems of the two organizations.

The April meeting was devoted to a discussion on the Boiler Rating Code, which was introduced by H. H. Angus at this meeting.

Pres. L. A. Harding was the guest at the May meeting, and he addressed the members and guests of the Ontario Chapter on Solar Radiation. The Ontario Chapter were highly honored by having President Harding as their guest.

The officers elected to serve the Ontario Chapter for the year are as follows:

President, H. J. Church.
Vice-President, H. S. Moore.
Secretary-Treasurer, H. R. Roth.
Board of Governors, H. J. Church, H. S. Moore, H. R. Roth, Barry Watson, W. R. Blackhall, J. S. Wood and W. Boddington.

Pittsburgh

The Program Committee of the Pittsburgh Chapter was fortunate in arranging for a number of outstanding speakers for the 1930 season. Many interesting and enlightening subjects were discussed.

W. S. Scott, heating engineer of the Westinghouse Electric & Mfg. Co., was the guest speaker at the January 1930 meeting. Mr. Scott spoke on Electrical Heating and Its Application.

Burning Fuels in Domestic Heaters was the topic of discussion at the February meeting, and was presented by Percy Nicholls, head of the Fuels Section of the U. S. Bureau of Mines Experiment Station. Mr. Nicholls devoted a greater part of his discussion to the Code for Rating Steam Heating Solid Fuel Hand-fired Boilers, with which he is thoroughly familiar.

At the March meeting, F. A. Gunther, Direct Control Valve Co., Pittsburgh, presented a very interesting discussion on the Economic Use of Steam in Modern Buildings. He also discussed the automatic temperature of control, the amount of steam consumption necessary for heating, and steam consumption as influenced by location of radiators.

For the April meeting the members and guests gathered at the Carnegie Institute of Technology so that they might enjoy a lecture on Psychrometry and Its Application to the Industrial Uses of Air by W. H. Carrier, President of Carrier Engineering Corp., Newark, N. J.

The officers for 1930 are as follows:

President, E. W. Stitt
Vice-President, Peter O'Neill
Secretary, W. W. Teague
Treasurer, G. M. Comstock
Board of Governors, H. Lee Moore, W. W. Stevenson and H. B. Orr

During the year, Mr. Stitt moved from the Pittsburgh district, and he was succeeded by W. W. Stevenson. S. E. Dibble was appointed to take Mr. Stevenson's place on the *Board of Governors.*

Fourth National Fuels Meeting Program

The program for the Fourth National Fuels Meeting of the *A. S. M. E.* which will be held in Chicago, February 11 to 13, 1931, is rapidly nearing the stage where papers and authors for each session may be announced to the Engineering World.

The Fuels Division of the *American Society of Mechanical Engineers,* which has sponsored these national events since their inauguration in 1927, is making an unusual effort to develop the type of program best fitted to parallel the sessions of the Mid-west Power Engineering Conference to be held during the same week.

Among the subjects included in the program in its present form are the following, each of which will be discussed by a prominent authority in the fields represented:

- Developments in Furnace Design
- Developments in Pulverized Fuel Firing
- Developments in the Small Stoker Field
- The Use of Refinery Waste Fuels
- The Use of Natural Gas and the Conversion of Coal Fired Furnaces to Gas
- Boiler Efficiencies as Effected by Types of Fuels
- Effect of Temperature on Rapidity of Combustion
- Heat Absorption as Affected by Furnace Wall Materials
- Selection and Co-ordination of Steam Generating Equipment
- Design and Performance of Large Boiler Units
- The Boiler Plant of the Future
- Changes in Character of Mid-West Coals and Stoker Performances
- Air Pollution

A. G. A. to Establish Testing Station on Pacific Coast

The *American Gas Association* is to establish an inspection station of its Testing Laboratory on the Pacific Coast, and it will be located at Los Angeles, California.

It was pointed out by members of the Laboratory Managing Committee that the cost of shipping equipment to the Testing Laboratory at Cleveland for approval is very high, particularly for Pacific Coast concerns. However, as it is not planned to establish an Approval Testing Station on the Pacific Coast, this item will be of consequence only when it is necessary for appliances to be returned for correction. Such additional expense, it is explained, undoubtedly could be avoided, at least in part, were appliances given a careful inspection before they are shipped to the Laboratory at Cleveland.

New York University Announces Evening Courses

Among the various engineering courses to be given in co-operation with the Extension Division of New York University, which have been announced by the Director of Engineering at the University, is the course devoted to heating and ventilating. This course is designed to augment the practical experience of those now employed in the design, erection, and operation of heating and ventilating systems, as well as to prepare those who are about to enter the field.

The curriculum for this course has been very ably prepared with the assistance of an advisory committee composed of A. A. Adler, W. H. Carrier, W. H. Driscoll, W. L. Durand, W. W. Hubbard, H. C. Meyer, Jr., T. S. Tenney and C. M. Wooley. It includes heating systems; chemistry of air, water, metals, and fuels with laboratory practice; theory of heat with laboratory work; principles of heating and ventilating design; application of heating and ventilating design; heating and ventilating testing; estimating principles and commercial practices; and mechanical and electrical equipment of buildings.

The other phases of the engineering industry to which special attention has been given are practical aviation, building construction and plumbing. These courses, together with the heating and ventilating course, are part of a three-year engineering program which leads to a certificate of proficiency, although they are not credited toward a degree.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the name of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 12 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by November 15, 1930, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

	REFERENCES	
CANDIDATES	*Proposers*	*Seconders*
Barnum, Willis E., Jr., Sales Engr., York Ice Mchy. Corp., Los Angeles, Calif.	W. S. Kilpatrick H. S. McClelland	C. S. Anderson H. B. Keeling
Bernstrom, Bert, Lakeside Co., Hermansville, Mich.	W. J. Uhlhorn L. L. Narowetz, Jr.	J. H. Watson C. P. Barry

Bunker, Kenneth Southworth, Union Tank & Pipe Co., Ltd., Los Angeles, Calif.	O. W. Ott C. S. Anderson	P. E. Bradfield A. R. Johnson
Campbell, E. K., Jr., Thermidaire Corp., Kansas City, Mo.	E. K. Campbell J. V. Martenis	N. W. Downes W. T. Jones
Cushman, Lester D., Mech. Engr., James H. Ritchie & Assoc., Boston, Mass.	J. S. Webb T. F. McCoy	G. H. Gleason C. R. Swaney
DeLancey, Ralph W., Sales Mgr. and Designing Engr., Real-Oil-Heat Burner Co., Winona, Minn.	R. W. Stockwell (*Non-member*) C. H. Graham (*Non-member*)	J. Bergstrom (*Non-member*)
Gerrish, Grenville B., Southern New England Manager, Kewanee Boiler Co., Inc., Boston, Mass.	W. T. Jones D. S. Boyden	Raymond Newcomb G. E. Olsen
Hashagen, John B., Sales Engr., Johnson Service Co., New York, N. Y.	F. E. W. Beebe Louis Sklenarik	J. A. Rice Kennedy Duff
Kennedy, Maron, Sales Engr., York Ice Mchy. Corp., Los Angeles, Calif.	W. S. Kilpatrick H. S. McClelland	C. S. Anderson H. B. Keeling
MacKinnon, Daniel A., Sales Engr., English & Lauer, Inc., Los Angeles, Calif.	Harrold English H. B. Lauer	E. H. Kendall J. S. Earhart
Rohlin, Karl W., Warren Webster & Co., Camden, N. J.	C. G. Binder Warren Webster, Jr.	Karl Rugart P. B. Stanger
Wallace, Bruce, Own Business, Newmarket, Auckland, New Zealand.	E. H. Rhodes—A. S. C. E. N. L. Paterson—I. C. E. M. K. Draffin—A. R. I. B. A.	A. S. Miller—A. V. C. L. E. Brooke—N. Z. I. A. G. Farrell—N. Z. I. A.

Candidates Elected

In past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

Fitts, Joseph C., Secy., Heating and Piping Contractors National Assn., New York, N. Y.

Peters, Herbert Henry, Manager of Branch, L. J. Mueller Furnace Co., Salt Lake City, Utah.

Ste. Marie, Gaston P., C. A. Dunham & Coy, Ltd., Montreal, Que., Canada.

Storch, Clemens A., Sales Engr., Johnson Service Co., Chicago, Ill.

Tulloss, Joseph C., Assoc. Mech. Engr., Construction Division, Office of Quartermaster General, Washington, D. C.

Winch, Franklin R., Consulting Engineer, 1031 S. Broadway, Los Angeles, Calif. (*Reinstatement*).

ASSOCIATES

Armstrong, John Alexander, Manager, A. M. Byers Co., Los Angeles, Calif.

Bynum, Otis William, Carrier Engineer Corp., Newark, N. J.

Gray, Richard Fulton, Dist. Repres., Iron Fireman Manufacturing Co., Cleveland, Ohio.

Oster, George Ray, American Air Filter Co., Louisville, Ky.

Pertak, William R., Allen and Billmyre Co., New York, N. Y.

Roth, Charles, Vice-President, International Exposition Co., New York, N. Y.

Swenson, John Edward, Minneapolis Gas Light Co., Minneapolis, Minn.

Thornton, Frank, Jr., Manager, Residence Engrg., Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.

Tupper, George B., Airmaster Corp., Chicago, Ill.

JUNIORS

Anderson, Edwin L., Carrier Engineering Corp., Newark, N. J.

Benson, John C., Carrier Engineering Corp., Newark, N. J.

Coburn, Luther F., Carrier Engineering Corp., Newark, N. J.

Darby, Marion Henry, Carrier Engineering Corp., Newark, N. J.

Dowling, Joseph Michael, Carrier Engineering Corp., Newark, N. J.

Edgar, James Struthers, Carrier Engineering Corp., Newark, N. J.

Gillham, John N., Carrier Engineering Corp., Newark, N. J.

Haskett, Joseph Elwood, Carrier Engineering Corp., Newark, N. J.

Mann, Lee Bernard, Carrier Engineering Corp., Newark, N. J.

Meffert, George H., Carrier Engineering Corp., Newark, N. J.

Paget, Francis King, Carrier Engineering Corp., Newark, N. J.

Powers, Lowell Gable, Carrier Engineering Corp., Newark, N. J.

Pulliam, William Gold, Carrier Engineering Corp, Newark, N. J.

Skidmore, John Gardner, Carrier Engineering Corp., Newark, N. J.

Smith, Frank H., Carrier Engineering Corp., Newark, N. J.

Van Meter, Benjamin F., Carrier Engineering Corp., Newark, N. J.

Walton, Louis A., Carrier Engineering Corp., Newark, N. J.

Ware, James Elliott, Jr., Carrier Engineering Corp., Newark, N. J.

Witmer, Charles Newcomer, Carrier Engineering Corp., Newark, N. J.

Heating·Piping and Air Conditioning EDITORIALS

"Use Your Head"

That system of maintenance is cheapest which results in an uninterrupted usage of equipment during its normal lifetime. Few executives or engineers will defend a system of maintenance which even infrequently interferes with production. Equipment which will not operate without delays should be replaced with other equipment, more carefully selected or designed, to prevent these delays. The whole machine age is based on continuous operation, and like an airplane flight across the ocean, no stop can be made for repairs.

Probably the one industry which learned the value of maintenance first was the railroads. They early learned that at all costs train failures must be avoided, and the well managed roads are governed accordingly.

That equipment is best, which when it meets the specifications for its use and can be secured at a reasonable cost, requires the least maintenance and the least care. No one would argue to the contrary.

A problem confronting many manufacturers today is for what length of time should equipment be purchased. Suppose a concern used tanks in its processes, that one type of tanks having a ten-year life cost a fixed amount, and that a second type having a twenty-year life cost twice as much. Mathematically the equation would look equal, but the wise and progressive concern would buy the first type realizing that old man "Obsolescence" is always just around the corner and has ruined many a business. Of course, if the second type lasting twice as long would cost one and one-half times as much as the first, then the plot thickens and there would be little choice. Good business dictates, however, to be careful of long commitments.

We were amused recently at some plated silver spoons, knives and forks offered for sale and guaranteed for one hundred years. Probably fifty years from the time of purchase both buyer and seller would be dead, all guarantees in writing would be lost, and most of the silverware gone in the garbage or have disappeared in some of the other natural channels. The problem of maintenance and obsolescence gets down to the same basis as the story of the golfer who stood over the ball saying, "Shall I use a niblick, mashie or midiron," and the little bird in a tree nearby chirped "Use your head." This was excellent advice.

Sound Insulation

The insulation of heat and sound in modern buildings presents an important problem. Since such insulation involves the nature of the building construction, it is not strange that engineers in these realms find it necessary to cooperate in securing an adequate solution. One evidence is the fact that the heating and air conditioning engineer is gradually widening his domain and now has definitely entered the field of sound proofing or sound absorption.

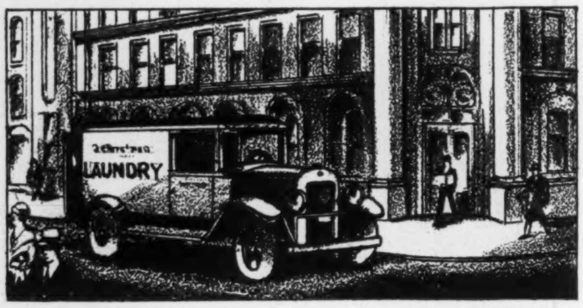
Noise is ever on the increase. We have developed the radio, and sound amplifiers of many kinds. We can make more noise now than at any previous time in the history of the world. Almost every new machine which contributes to our comfort adds its noise. Many of these noises are made in buildings, some of which are as reverberant as the sounding box of a violin. Even slight noises are often carried through walls, corridors, along studs or other solid members, through ducts and passageways, often in a most amazing manner.

Sound isolation is not merely for comfort, it is also a health measure. The National Safety Council reports that city noises are surely taking toll of human life. New York City, realizing the necessity of cutting down noise, has appointed a noise abatement committee. Their first report has just been published. Sound proofing is now demanded in apartments, hotels, hospitals, private and public schools, auditoriums, churches, industrial plants, office buildings and many other types of structures. The heating and air conditioning contractor and engineer must now know not merely how to design, specify and install heating and air conditioning equipment but he must also know how to do it so that it will not cause or transmit annoying sounds.

Some Observations on the

Air Conditioning of Laundries

"AIR Conditioning is a very important factor in the operation of a laundry, but one that is frequently disregarded by the management," says Walter H. Pierce, director of the department of engineering, *Laundry Owners National Association.* The problem should, of course, be studied and solved at the time the laundry building is constructed. However, there are many laundries operating in buildings not primarily designed for their use; hence, it is necessary in such cases to make the air conditioning system as practical as possible.



Air conditioning does not mean just creating a draft or agitating the air in a room or building. It means supplying fresh air in an adequate quantity at the right temperature and humidity.

Generally the moisture content or the humidity of the air is of greater importance than its temperature. A person can be healthful, active and feel comfortable in temperatures ranging from 65 to 80 F, provided the moisture content is in a proper ratio to the temperature.

Removing Excess Heat and Vapors

A laundry plant has a serious ventilating problem due to the presence of considerable heat given off by steam-heated equipment as well as the vapor given off by flatwork ironers, presses, washers, etc.

The first step in solving the problem of ventilation is to remove as much of the excess heat and vapor as possible. The vapor or heat rising from a flatwork ironer can be fairly well disposed of by installing a hood or canopy over the ironer. This hood should be connected to the outside by means of a duct. In some cases, the natural draft set up by the difference in temperature is sufficient to draw off the heat or vapor. In many cases, however, it is necessary to install a fan to create an induced draft. There are hoods manufactured by machinery companies that fit the various sized ironers. A simple hood of galvanized iron will serve the purpose very well. Such a hood should be large enough to cover both end rolls and it should be as wide as is the ironer.

The general scheme for introducing fresh air into the plant will depend upon the location of the plant both geographically and locally, and the type of building, whether single or multiple story.

Construction Determined by Climate

Plants located in a warm climate do not have to be as substantially built and are usually more open than those found in a cooler climate. Plants in warm sections usually need a cooling system to cool the air during the hot seasons.

The plants located in cool or cold climates must be built to withstand the rigorous winter weather. They have the problem of not only cooling the plant in the summer months but heating it in the winter.

Industrial Considerations

The location of a plant relative to other industrial institutions must also be considered. A plant located in a steel mill town, or one equal to it in amount of soot and dust deposits, must have an air conditioning system different from one located in a small town without mills, or in a residential section comparatively free from smoke and dust.

Ventilating Sash and Monitors

The single story plants located in clean sections of a town or city can usually be successfully ventilated by constructing the building with ventilating sash along the sides and a monitor construction on the roof. This provides for fresh air entering the building at the sides and warm air and vapor leaving through the monitor. Provisions should be made in plants located in cool sections for radiators under the ventilating sash for heating the fresh air supply during the cool season.

Cleaning the Air

The multiple story and single story plants situated in smoky and dusty localities should have an air conditioning system that will deliver clean air at the desired temperature and humidity. A satisfactory system for a laundry calls for a change of air fifteen times per hour or once every four minutes.

Relation to Outdoor Conditions

The relations between the most desirable mechanically maintained air conditions and the prevailing outdoor conditions have been definitely established, and are as follows:

Outdoors (Summer) Dry Bulb—F	Indoors		Relative Humidity %
	Dry Bulb—F	Wet Bulb—F	
95	80.0	65.2	49
90	78.0	64.5	50
85	76.5	64.0	51
80	75.0	63.5	53
75	73.5	63.0	57
70	72.0	62.5	60

PRACTICAL
PIPING
PROBLEMS

Tightening Flanged Joints on Hydraulic Piping
An Indicator for Underground Lines

by
W. H. WILSON

Indicator Valve for Pipe Line

Operating and maintenance requirements for main pipe lines call for shut-off valves to be equipped with a reliable indicator, to show whether the valve is in the open or the closed position. Exposed pipe lines, not affected by outside weather conditions, can generally be equipped with the outside screw and yoke type. Underground lines make use of the post indicator for operating valves below ground. The illustration shows the installation of a valve equipped with a post indicator for use on a pipe line for 100 pounds compressed air pressure.

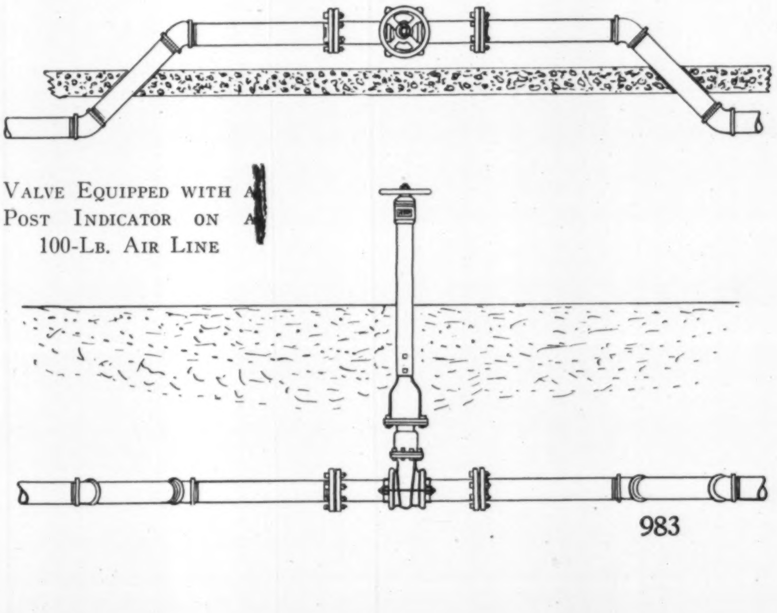
This pipe line was installed in a concrete tunnel and supplied air pressure to several buildings at a distance. The size of the tunnel did not permit room for the valve to be installed in the tunnel at the desired location.

An offset was made in the pipe line to allow a section of the pipe to be placed outside of the tunnel at the place where it was desired to locate the shut-off valve. A water gate valve, bell end type, for the regular water pipe, lead-calked joint, equipped with an indicator post flange, was used.

In order to make a serviceable joint and to prevent any possibility of its blowing out, flanged and spigot pieces were used in the bell ends of the valve. The hemp was put in the joint firmly and a generous amount of lead was securely calked in place. This calking was done while the valve was above ground, making it convenient to produce a good joint. As further security, a clamp of the split type was applied to the spigot pieces at the joint and both clamps held together with tie rods. The valve was then placed in position and connected with the flanged joints.

Screwed fittings were used for the remainder of the joints and after the piping was tested, the pipe, fittings, flanges and the valve were painted with a preservative coating, especial care being taken to see that it was applied where the pipe passed through the walls of the tunnel. The post indicator was set in position and the backfilling completed.

This arrangement permitted the valve to be operated above ground, provided a reliable indicator and the lead-calked joint, reinforced by clamps, prevented the air pressure from blowing out the joints.



Valve Equipped with A Post Indicator on A 100-Lb. Air Line

Tightening Flanged Joints

The tightening of the bolts in flanged joints is one of the important parts of the installation of good pipe work and piping equipment. It is easily understood that bolts are to be tightened evenly in order to give a correct bearing surface on the gasket faces. In hydraulic work, because of the higher pressures usually carried, heavy fittings and strong bolts are required. The tightening of these bolts should be given careful attention.

The illustration on the next page shows a connection between two hydraulic cylinders that is likely to require attention from the maintenance employes in order to keep it free from leaks developing as the joints remain in service.

These joints are rigid, as is usual with heavy hydraulic machinery. The joints are generally of the male and female type, which will require the elbows and the valve to be placed in position and then set in place on the ends of the cylinders. The bolts in the upper ends of the elbows where they join to the valve should be pulled up well before the final tightening is done to the bolts at the lower ends of the elbows, where they are joined to the cylinders.

By tightening the bolts in this way, the gasket faces will get an even bearing on the joints at the ends of the valve and the lower part of the elbows will be free to move slightly. This clearance is gotten from the bolt holes being slightly larger than the bolts and some clearance can be gained on the female opening in the joint at the lower end of the elbows where they are connected to the cylinders. In this case, making the female opening slightly larger, possibly a sixteenth of an inch, will allow the elbows to move slightly as the bolts of the joint at the contact of the elbows and the valve are tightened. The bolts in the joints at the lower end of the elbows, shown at the lower left-hand corner of the illustration, are tightened last. There is nothing in the way to prevent them drawing down into position and be free from exerting any strain on the other joints.

If the joints at the end of the elbows that